ELEMENTARY MACHINE DESIGN

HOFFMAN

A COURSE OF INSTRUCTION

IN

ELEMENTARY MACHINE DESIGN

ARRANGED FOR

STUDENTS OF THE JUNIOR CLASS
PURDUE UNIVERSITY
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BY

J. D. HOFFMAN, M. E.
Associate Professor in Engineering Design

PRESS OF BURT-TERRY-WILSON CO
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PREFACE.

In outlining this course of study in machine design it has been considered wise to depart from the customary methods employed in presenting the subject, i. e., that of dealing with a number of isolated problems that have no relation one to the other, and in its place to take up the study of completed machines, analyze their mechanisms and design the various details. To design any piece of machinery in an intelligent manner it is necessary that the person be acquainted with the conditions surrounding that piece in the machine, hence the first thing to do is to analyze the mechanism of the machine and find the magnitude and direction of the acting forces; this will give him some thread and meaning to his work. After a general plan has been laid for the work then each individual piece can be taken up, in its logical order and designed to meet the conditions. All parts that admit of theoretical analysis should be so treated and when such treatment is not apparent, or for some unknown cause can not be applied the parts may be proportioned from existing and parallel cases.

The following pages, are not presented as an exhaustive treatise on machine design but rather as a comprehensive analysis of typical machines, illustrating how such work would be handled.

In compiling the notes, frequent reference was made to the following standard works on Machine Design, to each of which the author is indebted for much valuable material. The student should make much use of these works throughout the Design Course.

The Constructor, Reuleaux.

Elements of Machine Design, Parts I and II, Unwin.

Machine Design, Parts I and II, Jones.

Machine Drawing and Design, Low and Bevis.

Mechanical Drawing and Machine Design, Reid and Reid.

Mechanical Engineers' Pocket Book. Kent.

Cambria and other Trade Books.

Elementary Machine Design

CHAPTER I

Materials Used in Machine Construction.

One of the first qualifications that the machine designer must have, is a thorough knowledge of the qualities of the materials ordinarily used in machine construction. It is therefore considered necessary that a brief discussion of these materials be given.

1. Iron Ore:—The basis in all the iron and steel production is the iron ore. The various ores are: Magnetite, Red Hematite, Brown Hematite and Ferrous Carbonate. These ores vary in color from a black to a yellowish brown and have theoretical percentages of pure iron of, from 72.4 to 48.3 respectively. Of the ores mentioned Red Hematite is of the most importance in the production of pig iron. It is found in abundance and contains about 70 per cent of pure iron.

2. Pig Iron:—Pig iron is obtained from a reduction of the iron ore in the blast furnace. The charge of the furnace is made up of fuel, ore and limestone or flux. The limestone forms a fusible slag with the silica of the ore and is floated off as a scum, while the free iron is taken off through the bottom of the furnace and carried in channels to open molds made in the sand on the floor. The metal when run into the small mold, is called a *pig*, and that in the channel a *sow*.

Pig iron contains a number of impurities, the principal ones being carbon, silicon, sulphur, phosphorus and manganese.

3. Carbon in iron is found in chemical combination and also in the free state as graphite. The percentage of carbon seldom exceeds 4 or 5 except in special combinations rich in manganese and chromium, where it is said to be found as high as 7. Combined carbon increases the tensile strength and the hardness of cast iron but diminishes its ductility. The amount of free carbon has no effect on the quality of the iron, but it indicates a metal having more ductility and less tensile strength than where the free carbon is less. The manipulation of the iron determines to a large degree in which of the two states the carbon will be; for example, a piece of iron that has cooled slowly from a melted state presents considerable free carbon, but the same piece of iron with the same sum total of carbon, when chilled or cooled suddenly, presents little if any free carbon. Increasing combined carbon generally reduces the silicon.

4. Silicon unites chemically with iron and is found in all blast furnace irons. Its presence in the charge has a tendency to reduce the amount of combined carbon in the iron, and to increase the graphite, thus softening the iron and making it more fluid. Silicon, alone in the iron, increases shrinkage and increases hardness, but by increasing silicon and consequently changing combined to free carbon, as would usually be the case, it reduces shrinkage, and softens the iron. The maximum strength of cast iron is probably obtained with 2 to 3 per cent. silicon.

5. Sulphur unites chemically with iron and is found in all blast furnace irons. Its presence is favorable to combined carbon and antagonistic to silicon. It increases shrinkage and above .1 per cent weakens the casting. Tests have been made that showed iron having small per cents of sulphur to be stronger than iron with little or no sulphur in it.

6. Phosphorous in small quantities is found in chemical combination with all blast furnace iron. It is not objectionable in small quantities as it makes the metal more fluid, and decreases shrinkage, but in quantities above 1 per cent it seriously weakens the iron without any corresponding benefit.

7. Manganese combines with iron in almost any proportion. It is originally found in *spiegeleisen* from 10 to 25 per cent, and in *ferro manganese* from 25 to 90 per cent. When iron containing manganese is remelted, the latter decreases by volatilization. Increasing manganese increases combined carbon, increases shrinkage and reduces silicon. It also increases the tensile strength and fluidity making the metal harder and less fusible.

8 The iron obtained by the blast furnace process is the basis of all the commercial irons and steels. The quality and composition of the iron from the various charges differ somewhat and it becomes necessary to assort the iron to suit the special requirements. The classification is usually as follows: *Bessemer* iron, used in the manufacture of Bessemer steel; *basic* iron, used in the basic process of steel manufacture; *mill* iron, used in the puddling furnace for the manufacture of wrought iron; *malleable* iron, used in making malleable iron castings; and *charcoal* iron and *foundry* iron, used for general utility castings and foundry work.

This classification of the iron may be accomplished in either one of two ways: by fracture, which is the more common way, or by chemical analysis, which is the more scientific way and which no doubt gives more satisfactory results.

9. Cast Iron:—The *charcoal* and *foundry* irons, usually called *pig irons,* are shipped throughout the country for use in the foundries as a basis for all gray iron and malleable castings. Pig iron is melted in a cupola with a certain percentage of scrap cast iron, and then poured into a mold, thus becoming a casting. Pure pig iron makes a casting that is very soft and easily machined, but one having a low tensile strength, hence the pig is usually mixed with other grades of cast iron for commercial use.

The process of remelting cast iron has the effect of burning out the free carbon and increasing the combined carbon and sulphur. The result of this process is an increased hardness and a more finely divided crystalline structure each time it is remelted. This mixing and remelting may be carried on through a number of stages from the *gray* colored pig iron with its open texture, to the *white* cast iron with its fine granular appearance.

The following summarizes the above statement:

Cast Iron 1½ to 4 per cent Carbon	Gray Cast Iron	Grades A	High in free carbon; gray color; low tensile strength; soft; fairly tough; when melted, is very fluid and has moderate shrinkage; used in castings requiring heavy machining and little strength.
		B	Medium in free carbon; light gray color; strong; easily machined; used mostly on machinery castings.
		C	Low in free carbon; white; crystalline; hard; brittle; when melted has moderate fluidity and heavy shrinkage; used on heavy machinery castings, castings subjected to excessive wear, and castings requiring little or no machine work, structural castings, etc.
	White Cast Iron		Very close, white granular appearance; stronger than gray iron but more brittle; carbon in chemical combination; used largely for conversion into wrought iron and steel.

10. *Gray cast iron* is the chief metal in machine construction because of its almost universal adaptability. It is more easily and more cheaply formed into intricate shapes than any other metal; it resists oxidation better than wrought iron or steel and it has a high compressive strength. It has, however, the following disadvantages: low tensile strength; cannot be riveted or welded; very brittle; very liable to hidden flaws and defects.

Cast iron may be *annealed* by heating to a light red and cooling very gradually. Annealing cast iron reduces its strength and increases its toughness.

11. The *shape* of a casting also affects its *strength.* It is generally known that the lines of strength in a casting, form while cooling. These lines are perpendicular to the face of the casting, and in an interior angle may be the cause of weakness. If a casting A, Fig. 1 should have forces applied in the direction of the arrows, the angle at *a* would be considered unsatisfactory. At *b*, however, owing to a better arrangement of the lines in cooling, the strength would be decidedly increased.

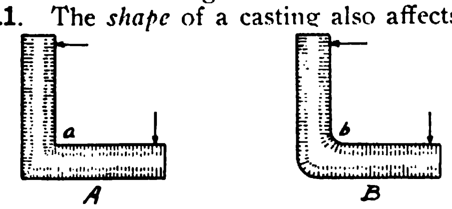
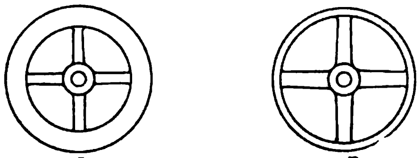


FIG. 1.

12. Cast iron *shrinks* about .01 of each linear dimension when cooling in the mold; this quality should be remembered by the designer wherever pattern sizes are specified.



A B

FIG. 2.

13. The designer should so *Proportion* a casting that the cooling in the mold will not throw an unnecessary strain upon any part. In A, Fig. 2, the rim of the pulley, being heavy, will set last, and in shrinking will cause a heavy compression in the arms, with a corresponding tension in the rim. If the arms are weak compared to the rim they may be thrown out of their plane. In B, the arms being heavier than the rim, they will set last and in shrinking will cause a compressive stress in the rim with a corresponding tensional stress in the arms. If the arms are stronger than the rim the latter will be thrown out of a circle, but if the rim is stronger, the arms will crack under any sudden blow; thus relieved, the rim again assumes its circular shape.

14. The *warpage* of a casting may be shown by Fig. 3. The portion *a b*, because of its relative thinness, sets first in a straight line and *c d*, cooling last, will draw *a b* away from the original position and cause the face to become convex. Other forms of distortion in castings from the same general causes might be shown but this one is considered sufficient. When it becomes a necessity to so proportion a casting that there will be evident distortion from warpage, the designer may anticipate this distortion by building up on the parts thrown out, so that the final casting may have the desired shape. The designer should not overlook the fact, however, that there will be a heavy internal stress in the metal whenever warpage exists. The preferred way would be to proportion the casting so the cooling strains would be neutralized.

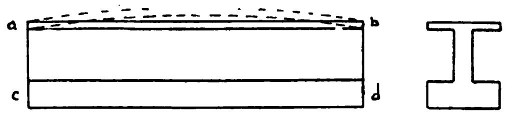


FIG. 3.

15. **Wrought Iron**:—*Mill* iron is placed in a refining or open hearth furnace and melted under a strong blast. Most of the impurities in the iron are burned out as it melts and it finally becomes a thick spongy mass at the bottom of the furnace. This is collected in the form of a *sponge ball* or *bloom* on the end of an iron bar and is taken to a squeezer where the slag or dirt is squeezed out of it. After this the bloom is subjected to repeated heatings, hammerings and weldings, until it is of a uniform texture, when it is finally drawn out under hammer or roll into the various shapes of the commercial bar iron.

Pure wrought iron contains no carbon. It is seldom produced in a pure state because of its extreme softness which renders it of little practical value in commercial work. Carbon is the hardening agent. and in some grades of wrought iron it may be found as high as .2 per cent. Where carbon is found in excess of this value the metal is classified as a mild steel.

16. The *quality* of the wrought iron depends upon, first, the removal of the carbon, and second, the care with which the bloom is worked up. The latter determines the *grain* or *fibre* of the metal The more thoroughly the bloom is worked up the more homogeneous will be the metal and the better the quality. Rolling and hammering iron increases its strength because it increases its density, hence the ultimate strength of wires and small bars is greater than that of larger bars. When wrought iron is rolled cold under heavy pressure it obtains a smooth polished surface and is greatly increased in strength.

Wrought iron is tough and *elongates* about .0001 of its length for each ton of load per square inch of section up to the limit of elasticity: beyond this point it elongates much more, the greater part of the elongation being permanent. When a bar is broken under tension it usually draws out at the point of fracture. This reduction of area is sometimes used as a measure of the fitness of the material to perform certain duties. In comparison with cast iron it is more satisfactory under tensile and transverse stress, but less satisfactory under compressional stress. It cannot be melted and run into molds like cast iron but it can be welded and forged into any desired shape.

Wrought iron cannot be tempered like steel but it may be *case hardened* by heating to a white heat and covering the surface with cyanide of potassium or ferro-cyanide of potassium and dipping in water. The cyanide slightly carbonizes the surface of the metal and gives it a quality about the same as steel. The chilling of the steely surface by the water, then, actually tempers it.

17. Malleable Cast Iron:—Malleable castings are obtained in two ways. The first and oldest method is by making a cast iron casting from white or mottled charcoal iron in an ordinary mold, and then subjecting this casting to a high heat in the presence of iron oxide. The carbon in the surface of the casting combines with the oxide of iron which surrounds it and leaves the surface of the metal with much the same qualities as wrought iron. In the process of decarbonization the castings are packed in cast iron or wrought iron boxes with red hematite ore or peroxide of manganese, and piled in a furnace; they are then kept at a bright red heat for several days after which they are allowed to

cool slowly. The second method is by the open hearth process. Malleable pig iron is melted in the open hearth furnace under blast. The process of decarbonization is carried on until the metal is of about the quality of semi-steel, when it is run into molds. Castings made in this way are of a more uniform texture and are stronger than those made by the first process.

Malleable castings are soft, tough, strong, flexible, and can be easily machined. Good specimens may be bent double, but could scarcely be bent back again.

18. To make a good malleable casting by the first process the *shape* should be such as to permit the greatest amount of surface in contact with the iron oxide. A thin rectangular, star, or ribbed section is better than a square or round section. The effect of decarbonizing is to produce a soft external malleable shell around a hard brittle core of cast iron. The thinner the section, the greater will be the percentage of malleable iron as compared to the cast iron, and the more tough and flexible will be the casting.

Malleable castings find their greatest usefulness in agricultural machinery, wind mills, small machine parts, ornamental work, and car and locomotive work.

19. Steel:—Steel is a chemical combination of iron and carbon and may be made by any one of three processes: the *crucible*, the *Bessemer* and the *open hearth* process.

20. *Crucible Steel,* or cast steel is produced by remelting blister steel (pure wrought iron bars heated in contact with charcoal or carbon until it absorbs a certain percentage of carbon) in a crucible and then pouring in molds. It is also produced by melting pure wrought iron in a crucible with enough charcoal and cast iron to introduce the required amount of carbon.

This grade of steel contains from .4 to 1.5 per cent of carbon, .1 to .75 per cent of manganese, .1 to .2 per cent of silicon, .01 to .02 per cent each of sulphur and phosphorus, and is used in making springs, cutlery, machine tools, and such machine parts as require hardening. The best all around tool steel has about 1 per cent carbon. Razors, lathe tools, drills, etc., have 1 to 1.5 per cent.

21. *Bessemer Steel* is produced in a Bessemer Converter by forcing a powerful blast of air through melted Bessemer pig. When most of the carbon is burned out a small quantity of cast iron containing a known amount of carbon is added to bring the carbon of the mixture up to a definite amount. This when thoroughly mixed is poured into ingot molds and becomes the basis for some of the cheaper grades of steel, such as, rails, nails, light shafting, merchant bar and some grades of rolled plate.

22. *Open Hearth* or *Siemens-Martin* steel is produced by melting a certain amount of basic pig iron with wrought iron or Bessemer steel scrap. The composition and manufacture vary somewhat according to the available scrap, the latter being replaced occasionally by iron ore having a known percentage of carbon. Open hearth steel is used in structural work, forgings, car axles and the better grades of steel plate.

23. All soft or mild steels are either Bessemer or Open Hearth. The former is produced more cheaply but the quality is sometimes inferior. On account of the cheapness of both these steels they have replaced wrought iron in most commercial work.

Mild steels generally contain low percentages of carbon, say from .02 to .06 per cent, and approach wrought iron in workability.

The *Quality* of steel is generally considered from the standpoint of the amount of carbon in it. Mild steel is soft, flexible, easily forged or machined and cannot be tempered. Cast steel, or steel high in carbon, is hard, rigid, not easily forged or machined and is readily tempered by heating and suddenly quenching in water. Exceptions to this are some of the self-hardening steels, as Mushet's steel or Hadfield's manganese steel, which harden by heating and cooling slowly in the air. All steels have a higher tensile and compressive strength than wrought iron or cast iron.

Mild steel may be somewhat hardened by hammering and rolling, as would be true of any soft metal, like wrought iron. This is due to the increased density of the surface material from the hammering or the rolling.

Steel is *annealed* by heating to a low red heat and cooling very gradually in a bath of dust, ashes or lime. High carbon steel should never be heated above a *low red heat.*

The *fracture* of steel presents a surface that is white, crystalline and fine grained. It has no fibre and usually gives a flat break.

24. Steel Castings:—Steel castings take the place of cast iron castings where great strength is required. The quality of the steel is varied to suit the requirements. Castings having about .1 per cent carbon are soft, tough and ductile; while those having .75 per cent carbon are very strong and rigid.. Ordinary steel castings contain from .2 to .5 per cent carbon. *Silicon* varies with the carbon in steel castings, from .1 to .4 per cent; *Manganese* from .5 to 1 per cent; *Phosphorus* .5; and *Sulphur* from .025 to .05 per cent.

Steel castings are most often produced by the open hearth process although they may be made by

both the Bessemer and the crucible process. They are used for such work as gears, hydraulic cylinders, engine frames and parts, locomotive driving wheel centers, large rolls, rolling mill spindles, coupling boxes, hammer heads and dies and ship and railroad castings.

Steel castings are more difficult to make than cast iron castings because of the unusually heavy shrinkage. They are frequently honeycombed and lack homogeneity, consequently a steel casting is usually made much larger than the size of the finished product to allow for sufficient machining. Castings high in carbon are less liable to be honeycombed than low carbon castings.

25. Alloys of Steel:—Some of the principal alloys of steel are Manganese, Nickel, Chromium, Tungsten and Aluminum.

26. *Manganese Steel.*—Manganese in steel has little effect below 1.5 per cent, but from 1.5 to 6 per cent the strength and ductility decrease and the hardness increases. At 6 per cent it is very brittle. From 6 to 14 per cent the strength and ductility again increases. The ductility is also brought out by sudden cooling. The best results with this steel are obtained at about 14 per cent. Carbon is present in about 1 per cent. Manganese steel is free from blow holes, welds with difficulty, cannot be softened, and combines extreme hardness and toughness. The later quality is one not found in any other metal and on account of it makes this steel very valuable for certain kinds of work. It is used for pins in elevator buckets, chain elevator links, jaws and plates on crushing machinery, safes, car wheels, axles, etc. The *ultimate tensile strength* of the best manganese steel is about 140,000 pounds per square inch.

27. *Nickel Steel* is made by the open hearth process and is used principally for armor plates. It is also very often used in commercial forgings and castings requiring very great strength and ductility. Nickel combines with steel having .25 to .4 per cent carbon, between the limits of 2.5 and 4 per cent. Nickel steel tubes are made containing 30 per cent nickel. It does not crack readily, has a high elastic limit ,is non-corrodible, is only slightly magnetic and seems to combine the toughness of raw hide with the strength of steel. Nickel steel is about 50 per cent stronger than ordinary steel having the same percentage of carbon.

28. *Chrome Steel.*—When 2 to 4 per cent of chromium is added to steel containing 1 to 2 per cent carbon, the steel becomes very hard and is able to resist severe shocks. Chromium is often added to nickel steel, making nickel-chromium-steel. Its principal use is in the production of armor plate and projectiles.

29. *Tungsten Steel* or *Mushet Steel* is a self hardening steel, i. e. will harden by heating to a red heat and cooling in air. This is one of the important steel alloys. The chemical compositions according to Mr. F. Reiser in "Stahl and Eisen," January 15, 1903, are, in per cents, about as follows: tungsten 5, manganese, 2.5, carbon, 2, chromium 0.5, and silicon 1.3.

Tungsten Steel is very hard and when hot becomes very brittle. It can only be worked between an orange and a bright orange heat, and then with extreme care. It is used chiefly for machine tools and since it is so hard that it cannot be worked cold, it is generally produced at the mill in standard sizes which require only grinding before using.

Some tests of tungsten steel by Styffe showed an ultimate tensile strength of 172,000 pounds per square inch.

30. *Aluminum* combines with steel in any proportion up to 15 per cent. It has no very important action upon the mechanical properties of steels when the percentage is low. Above 2 or 3 per cent it causes brittleness in the metal.

31. *Molybdenum Steel.*—The principal use of this steel is in tools. It is also used in large cranks, propeller shafts, gun barrels, boiler plates and wires. Molybdenum increases the elongation of steel very greatly. This is of special importance in the production of wires. It is usually found in combination with the nickel steels.

32. *High Speed Steels.*—The exact chemical compositions of high speed steels are unknown, except to the makers. According to Prof. L. P. Breckinridge in Bulletin No. 2, University of Illinois, on "Tests of High Speed Tool Steels on Cast Iron," the following elements are found in varying quantities: Carbon, tungsten, chromium, manganese, molybdenum and titanium; the Taylor-White steel having as high as 12 per cent tungsten and 4 per cent chromium.

When these steels were first produced, they were supplied to the trade in an unannealed state, but now they are usually annealed. The advantage of the later over the former is that it hardens better and is less liable to fail from internal stresses set up by the hammering and rolling process. Annealed stock may be nicked and broken, or it may be forged in an ordinary forge fire between a bright red and a bright yellow heat. It should never be hammered below a bright red heat. To *harden* this steel, it should be heated until it approaches a clear white heat, or at least until the flux begins to run on its surface, and then it should be cooled in an air blast, or quenched in a bath of lead or oil.

The latter is in many cases preferred since it makes a tool that is tougher and one that will resist greater shocks.

In the use of high speed steels a great increase in the work output can not be expected so long as the steel is used in the old design of machines. It is possible to increase the cutting speeds of metals as high as 300 or 400 per cent above that of the carbon steel, and as high as 100 per cent above the so-called self-hardening steel. It is possible also, to multiply the size of the cut as many times. Under such conditions it is apparent that machines designed for carbon or self-hardening steels will need redesigning before they can be successfully forced to the full capacity of the high speed tool.

33. Copper:—Pure copper is red in color, is ductile and malleable and can be forged hot or cold. It is cast in blocks and is then drawn out under hammers or rolls. Hammering and rolling increases its strength and brittleness. The latter quality may be removed, however, by annealing. Copper is anealed by heating and suddenly cooling. It may be hardened by heating and cooling slowly. In these particulars it is the opposite of the other metals. Hard drawn wire has about three times the tensile strength of cast copper, and is nearly equal to that of mild steel. When hard drawn copper is annealed it loses about 25 per cent of its strength.

It is difficult to get a sound copper casting. Good copper castings may be made somewhat easier by the addition of 1 to 3 per cent phosphorus.

34. Copper Alloys:—There are a great number of copper alloys. These alloys vary slightly in composition, but are known under the general names of *bronzes* or *brasses*. Of the bronzes there are compositions known as *Phosphor bronze*, *Aluminum bronze*, *Silicon bronze*, and *Manganese bronze*. Of the brasses there are those known as, *high brass* and *low brass*.

Formulas representing each of the above are as follows:

Phosphor bronze, Copper 89%, Phosphorus 1%, Tin 10%.

Aluminum bronze, Copper 90%, Aluminum 10%.

Silicon bronze, Copper 96%, Silicon 4%.

Manganese bronze, Copper 67½%, Manganese 18%, Zinc 13%, Silicon 0.5% and Aluminum 1%.

The following composition, usually termed bronze but sometimes called *Gun Metal*, is a very common one.

Copper 90%, Tin 10%.

In the brasses, the color usually designates the quality; high brass, yellow; low brass, red. The first shows low copper and high zinc or tin, and the second shows high copper and low zinc or tin.

High brass (ornamental work), Copper 65%, Zinc 33%, Lead 2%.

Low brass (bearing ,etc.), Copper 80%, Zinc 10%, Tin 5%, Lead 5%.

35. Babbitt Metal:—Some journal boxes, loose pulleys and other friction surfaces that are not considered of the highest order, are lined with babbitt metal, instead of bronze or brass. This makes a cheap and a fairly substantial bearing. Babbitt metal is the cheapest anti-friction metal known, and always contains two or more of the following ingredients: Tin, Copper, Zinc, Antimony, and Lead. It is probable that the original formula for this metal was about 90% Tin, 3% Copper, 7% Antimony.

Copper is not always used because it renders the bearing more liable to friction. Some of the cheapest grades of babbitt metal have a large percentage of Lead. This is usually mixed with Tin or Zinc. The ordinary grades sold in the market have a composition of about 45% Tin, 2% Copper, 13% Antimony, 40% Lead.

CHAPTER II

Notes and Formulas in Elementary Machine Design.

36. Machine Design:—In taking up the study of the design of machine parts our attention is called to the *Concrete* Side of Mechanism and Mechanics; *Machine Design* is an analysis of the definite forces acting upon pieces of material which must in turn be designed of sufficient size to resist these forces.

37. Resolution of Forces:—It will be assumed in this text that the student is familiar with the elementary work in *forces* and that a suggestion to review this topic and become thoroughly familiar with it will be all that is necessary here. Design No. 1 is calculated chiefly from the composition and resolution of forces hence this topic should be carefully considered. Such a review will contain in detail the composition and resolution of forces as given in, Reuleaux, "The Constructor," pages 26-38; Church, "Mechanics of Engineering," pages 1-18; Low and Bevis, "Machine Design," pages 18-20.

In the above be careful to thoroughly understand and be able to apply the *parallelogram,* the *triangle* and the *polygon* of forces.

The following general statements concerning forces should be well understood:

(1). Any force has a definite effect on a body no matter if that body is in motion or at rest.

(2). Any number of forces acting on a body will cause it to remain at rest, or to move in the direction of the resultant of the forces.

Forces act in one of five ways relative to each other: First, in the same straight line, and in the same direction, in which case we have the sum of the forces. Illustration, a ship propelled down stream under the action of the current and the engines; Second, in the same straight line and in opposite directions in which case we have the difference of the forces. Illustration, the ship propelled up stream; Third, in separate lines that intersect, in which case we have the resultant of the forces,as the ship crossing the stream under the action of her engines, and moving down stream by the current; Fourth, parallel forces acting in different lines, and in the same, or in opposite directions; Fifth, forces that are not parallel and do not intersect.

Conditions one, two and three are comprehended in the above while the fourth and fifth conditions bring in the idea of moments, a subject which will occupy the leading thought in Design No. 1, and should be considered by all as one of the fundamental principles of machine design.

38. Moment of a Force:—The moment of a force is *the product of the force and its lever arm.* If the force be measured in pounds and the lever arm in feet the moment will be in foot pounds. If the force be in pounds and the arm in inches the moment will be inch pounds. In most machine design calculations the latter would be used. When a system of forces acting on a body is in equilibrium, the sum of the moments acting in one direction about any given axis is equal to the sum of all the moments acting in the opposite direction about the same axis.

Illustration.—Given an imaginary beam having no weight and supporting a weight W as shown in Fig. 4. Find the reactions at A and B. Taking moments, first about B, and then about A, we have,

$$W\,b = A\,(a + b)$$

$$A = \frac{W\,b}{a+b} \qquad (1)$$

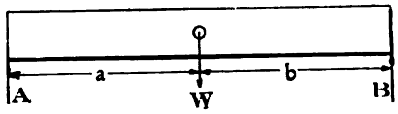


FIG. 4.

$$W\,a = B\,(a + b)$$

$$B = \frac{W\,a}{a+b} \qquad (2)$$

Also note that $A + B = W.$ (3)

Equation (3) combined with either (2) or (1) gives material for the complete solution of the beam.

Illustration 2.—Taking moments about A. Fig. 5, we have

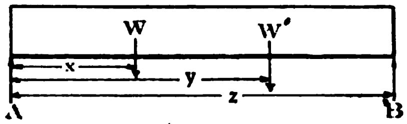


Fig. 5.

$$B\,z = Wx + W'y$$

$$B = \frac{Wx + W'y}{z}$$

from which if we remember that $W + W' = A + B$ we can get the solution of the beam.

Note:—If in the above the beam had weight, the bending moment due to this weight would in some cases be taken into account. In most problems in Machine Design the forces acting upon any beam are so much greater than the weight of the beam that the latter is disregarded.

In every exact calculation, however, the weight of the beam should be considered. It will be desirable for the student to bear this in mind when the subject is discussed more fully later.

39. Load:—A load is a combination of external forces acting on a piece of a structure, or machine. Illustrations: A weight carried by a rope; a weight supported on a beam; power transmitted through shafting or belting, etc., etc.

Loads are classified according to action as *dead load, live load, and shock*. They are also classified as to distribution, as *concentrated* and *distributed*.

A *dead* load is one that is applied slowly and remains constant. A *live* load is one that is continually changing, but is not subjected to any sudden applications. *Shocks* are due to sudden applications and withdrawals usually alternately in opposite directions. The first and second can well be represented as the dead weight of the bridge, and the live load of the train passing over it. The third can be represented as the load on the piston rod, or the connecting rod of an engine. A *concentrated* load is one that is applied at one point and a *distributed* load is covered more or less evenly over the surface. Distributed loads are commonly *uniformly* distributed, as for example, the load due to the floor of a bridge, also the uniform weight of the floor supports.

40. Effect of a Load:—When a load is applied to a body it produces a change of form, or strain, in that body. It also produces a corresponding stress on the fibres of the material. The change of form then is known as *strain,* and is usually expressed as a certain amount per unit of length.

That internal force which is called into play to resist this deformation is called the *stress,* and is expressed in pounds per square inch. In machine design we are usually concerned with the stress and less often have to consider the strain.

41. Kinds of Stresses:—There are three kinds of simple stresses: *Tensional,* tending to elongate *Compressional,* tending to shorten and, *Shearing,* tending to cut the material across the grain, or fibres. The following equation gives the relation existing between the load and the stress:

$$W = f\,A. \qquad (4)$$

where W = load applied in pounds,
f = stress on the fibres in #□".
A = area of the section of the material, or the "resistance of the section."

To illustrate the meaning of tension compression and shear; we have first a rod in tension supporting a weight Fig. 6. (A) where

$$W = f\,A.$$
$$3{,}000 = .7854\,f$$
$$f = \frac{3{,}000}{.7854} = 3819.7 \text{ \#□''}.$$

Second, a block under pressure. (B) where

$$W = f\,A.$$
$$3{,}000 = 4\,f.$$
$$f = \frac{3{,}000}{4} = 750 \text{ \#□''}.$$

A bolt or pin under cross strain. (C) where

$$W = f\,A.$$
$$3{,}000 = .7854\,f$$
$$f = \frac{3{,}000}{.7854} = 3819.7 \text{ \#□''}.$$

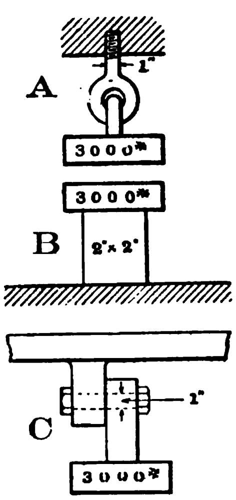


Fig. 6.

The following definitions should be kept well in mind:

The *ultimate stress* in any piece of material is the stress in pounds per square inch that the fibres are subjected to at the point of rupture. This factor differs in all materials.

The *Working Stress* is that stress in pounds per square inch to which a piece is to be subjected when it is a part of the machine. This is usually designated,

$$f = \frac{\textit{ultimate stress}}{\textit{factor of safety.}}$$

42. **Factor of Safety**:—The *Factor of Safety* is the ratio of the ultimate stress to the working stress. This is assumed by the designer from his knowledge of the kind of work the machine will perform To illustrate: In machine construction every piece sustains some sort of a load. If this load is a dead load the piece can be used under a greater fibre stress than if it be subjected to a shock. Consequently the factor of safety will be smaller. Conversely, if the piece is under shock the factor of safety should be taken large so the working stress would be low.

It is sometimes confusing, for the person just taking up the subject of Machine Design, to understand why the factor of safety should vary through such wide limits in the same material when used under different conditions. All materials, when subjected to continued usage, become weaker and the ultimate strength, after much service, is less than it was at first. This is more noticeable in materials subjected to live loads and shocks than in those subjected to dead loads. If the ultimate strength of new metal be taken as *one*, then subjecting this material continuously, for a considerable time to a live load with stress of one kind only, the ultimate strength may fall to about *one-half;* also if subjected to a live load of equal and opposite stresses, the ultimate strength may fall to *one-third.* Looking at the question from this standpoint, we see that the ultimate strength of the metal is really the variable factor.

TABLE I.—FACTORS OF SAFETY.

MATERIAL	DEAD LOAD	LIVE OR VARYING LOADS PRODUCING		
		Stress of one kind only.	Equal alternate stresses of different kinds.	Varying loads and shocks.
Cast Iron....................	4	6	10	15
Wrought Iron and Steel......	3	5	8	12

In designing any machine part for tension, compression or shear, it is first necessary to know the kind and amount of load which the part is to sustain. It is then necessary to know from the character of the material involved how many pounds per square inch it will stand before breaking, and assuming a factor of safety determine the working stress, or fibre stress f in the formula $W = f\,A$. This gives the area of the piece necessary to fill the conditions.

43. Table II which follows gives accepted values for the fibre stresses of the materials commonly used in machine design. It will be noticed that the figures given in some cases vary through wide limits; this is due to the difference in texture between different pieces of the same class of material. The designer should take especial notice of this in making his selection.

TABLE II.—Strengths of Materials Most Common in Engineering Construction.

MATERIAL	Per Cent of Carbon	Tensile Strength		Compressive Strength		Shearing Strength	Flexure	Modulus of Elasticity		Shock Re-sist'g pow'r	Methods of Shaping for Use
		Elastic Limit	Ultimate	Elastic Limit	Ultimate	Ultimate	Stress in Outer Fibre	Tension	Shear		
Cast Iron		No Elastic Limit	10000 to 35000 aver. 20000		50000 to 140000 aver. 90000	12000 to 25000	Ultimate 30000 to 54000 aver. 42000	16000000	7000000	Low	Casting.
Steel Castings		29000	47000 38000	29000	125000			20000000 to 30000000 av 25000000		High	Casting.
Malleable Cast Iron			35000			42000	Ultimate 70000			High	Casting.
Crucible or Tool Steel		65000	120000	65000				35000000	13000000	Medium	Rolling, forging and wire drawing.
Common Wrought Iron		22000	40000	22000	48000	35000		28000000	10000000	High	
High Grade Wrought Iron		28000	56000	28000	50000	45000	Elas. Lim. 50000	28000000	10000000	High	Rolling, forging and wire drawing.
Bessemer and Open Hearth Steel, also called Machinery Steel, Mild Steel and Ingot Steel	0.15 0.20 0.50 0.70 0.80 0.96	42000 45000 48000 53000 57000 68000	63000 65000 80000 89000 103000 115000	39000 40000 47000 53000 63000 71000		48000 50000 57000 60000 68000 83000		30000000	10000000	High	Rolling, forging and wire drawing.
Nickel Steel, Oil Tempered		60000	90000								
Nickel Steel, Annealed		45000	80000								
Brass			25000	12000	10000			9000000			Casting, rolling, forging and wire drawing.
Bronze			25000 to 50000 aver. 35000								Casting, rolling, forging and wire drawing.
Copper			20000 cast 60000 wire		58000	20000 to 30000		15000000			Casting, rolling, forging and wire drawing
Babbitt Metal											Casting.

44. Columns:—Any piece of material in compression, having a length of 15 or more times its least cross sectional dimension, is called a column or pillar. Ordinary rules for compression may not apply in such a case.

The strength of a column varies almost inversely as the square of its length, i. e. if a column is designed to sustain a load of 10,000 pounds, and its length be increased twice, it will carry approximately 10,000 ÷ 4 = 2500 pounds safely, providing the cross sectional areas remain the same.

The *condition of the ends* of a column affect its capacity to resist buckling. Fig. 7, *A* is *square ended; B* is *pin* and *square ended,* and *C* is *round ended. A* may be either flat, as shown, or fixed, as in *D, E* and *F. B* and *C* may take the forms of *G, H* and *I.* Flat ends should be well fitted, otherwise they should be classified as round.

When the length of a compression member is more than 15 or 20 times its least cross sectional dimension, it is well to investigate for buckling. The following formulas by *Rankine* give the breaking loads for *A, B,* and *C* and apply fairly well to columns of any length.

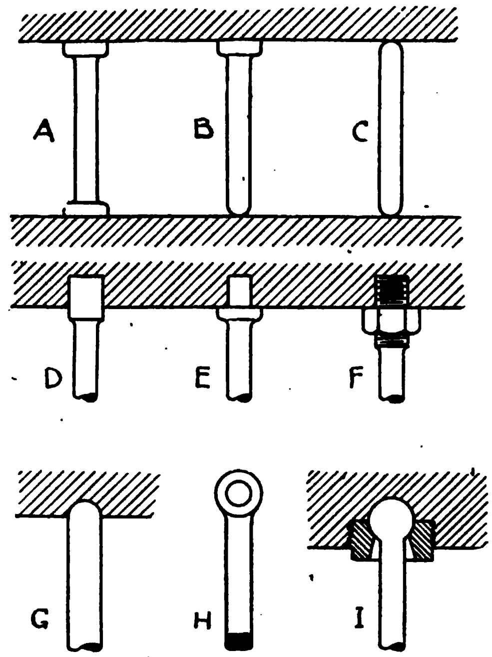


FIG. 7.

A. Square ends. $$P = \frac{CF}{1 + a\frac{l^2}{k^2}} \qquad (5)$$

B. Pin and square. $$P' = \frac{CF}{1 + \frac{16}{9}a\frac{l^2}{k^2}} \qquad (6)$$

C. Round. $$P'' = \frac{CF}{1 + 4a\frac{l^2}{k^2}} \qquad (7)$$

Where *P, P'* and *P''* are the breaking loads (afterwards to be reduced by the factors of safety); *F,* the total area of the section; *C,* the modulus of crushing (ultimate unit crushing strength); *a,* a constant, given in the following table; *l,* the length of the column in inches; and *k,* the radius of gyration of the section, in inches.

TABLE III.

Solid Sections	Cast Iron	Wrought Iron	Mild Steel	Hard Steel
C, in pounds per sq. in. =	80000	36000	67000	114000
a =	$\frac{1}{6400}$	$\frac{1}{36000}$	$\frac{1}{22000}$	$\frac{1}{14400}$

For columns having a length of, say, 150 times the least radius of gyration, the following, known as *Eulers* formulas will give satisfactory results.

$$\text{A. } P = 4\,E\,I\,\pi^2 \div l^2 \qquad (8)$$
$$\text{B. } P' = \tfrac{9}{4}\,E\,I'\,\pi^2 \div l^2 \qquad (9)$$
$$\text{C. } P'' = E\,I''\,\pi^2 \div l^2 \qquad (10)$$

Where *P, P', P''* and *l* are as stated above: *E,* the modulus of elasticity; and *I, I'* and *I'',* the moments of inertia of the section. From these formulas we have, with the same cross-section in each, if the strength of *C* = 1, then *B* = 9 ÷ 4. Also, for any definite load, if the length of *C* = 1, then the length of *B* = 3 ÷ 2 and *A* = 2, for columns of the same rigidity.

45. *Connecting rods* usually have a rectangular section, Fig. 8. Such a rod is a combination of A and C. It can be shown, as follows, that the height h should be twice the breadth b to be equally strong to resist buckling in either plane. To fulfil this condition $P = P''$ or $4\ E\ I\ \pi^2 \div l^2 = E\ I''\ \pi^2 \div l^2$. $4\ I = I''$. then $4\ b^3\ h \div 12 = b\ h^3 \div 12$ and $h = 2\ b$.

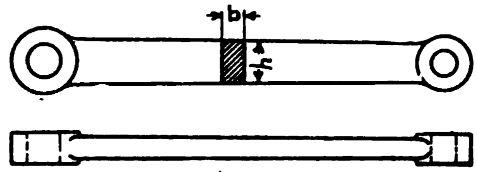


FIG. 8.

In applying the column formulas to machine parts, it is well to first design the parts to resist compression, and then, see if they are rigid enough to resist buckling.

Application.—A sliding ram is connected to the driving shaft of a machine by a rectangular sectioned, mild steel connecting rod, 36 inches long. This rod carries 100,000 pounds compressive stress. How large must be the section of the rod to resist compression and buckling?

Assuming this to be a slow moving machine, the safe compressive stress may be taken at 10,000 pounds per square inch. The area of the section then, is 10 square inches. With $h = 2\ b$, the section of the rod will be 2.25×4.5 inches. Now if the rod is equally strong in both planes, either formula A, or C (Rankine) may be applied. Taking the latter we have,

$$P' = \frac{CF}{1 + 4a\frac{l^2}{k^2}} = \frac{67000 \times 10}{1 + \frac{4}{22000}\frac{(36)^2}{1.69}} = 588094 \text{ pounds.}$$

This shows that the rod has a factor of safety of $588{,}094 \div 100{,}000 = 5.8$ for buckling, and 6.7 for compression.

46. **Bending**:—Another kind of strain and stress that is very common in machine construction is *Bending*. This deals with the moments of forces rather than with simple pressures, and our formula is:

$$M = f\ Z \tag{5}$$

where M = bending moment

f = fibre stress in pounds per square inch

$Z = \begin{cases} \text{Modulus of the section.} \\ \text{Resistance of the section,} \\ \text{or section factor.} \end{cases}$

This formula will be continually used and should be well understood. To make the explanation clear, assume a cantilever beam as in Fig. 9, projecting from a wall and acted upon by a weight W a distance l inches from the point of support. It is evident that the beam will be under a strain as shown by the dotted lines, with the upper fibres in tension, the lower fibres in compression and the neutral axis free from stress of any kind. It will also be understood that the stress on the fibres at any section $x\ y$ will be in a direct ratio to the distance from the neutral axis, consequently, it will follow that certain portions of the section will be under more stress than other portions. From this point of view the resistance of the section will not be in proportion to the area of the section, but to some other factor (Z) called the *modulus of the section*, or *section factor*. The modulus is found by dividing the moment of inertia of the section by the distance from the center of gravity to the outer-most fibre. (See Church's Mechanics, Par. 90) Z will be different for different shaped sections. In the reference as given the moment of inertia of a rectangular section about its gravity axis, is $b\ h^3 \div 12$. The distance from the gravity axis to the outermost fibre is $h \div 2$ where h is the height of the beam section, consequently, $2b\ h^3 \div 12\ h = b\ h^2 \div 6$ which is the value of Z in the formula, if the beam has a rectangular section.

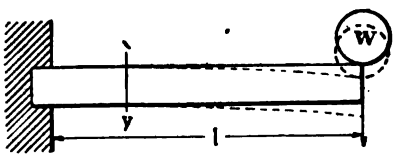


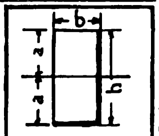
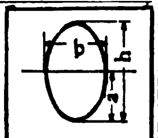
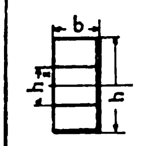
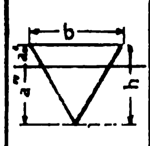
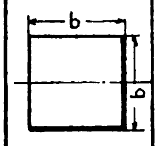
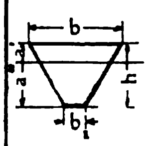
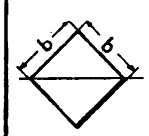
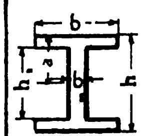
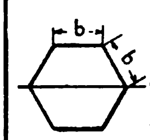
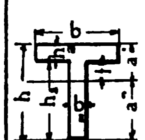
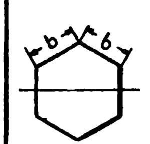
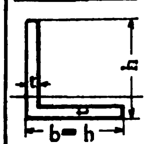
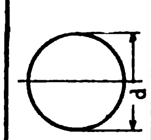
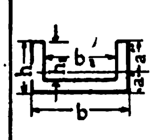
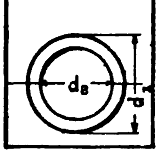
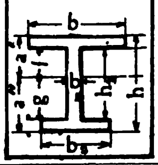
FIG. 9.

Suppose the beam is circular in section, the modulus from Par. 91, Church, becomes

$$\frac{\pi r^4}{4r} = \frac{\pi}{4}\left(\frac{d}{2}\right)^3 = \frac{\pi d^3}{32}$$

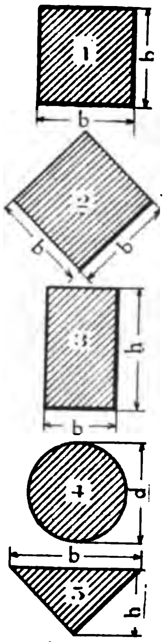
47. No attempt will be made in this work to figure out the values of the modulus Z for different sections since this is the province of the mechanics, but values of Z are given in the section table IV following for all common forms of sections.

SECTION TABLE IV. Bending.

Section	Modulus	Area	Section	Modulus	Area
	$\frac{bh^2}{6}$	bh		$\frac{\pi}{32} bh^2$	$\frac{\pi bh}{4}$
	$\frac{b(h^3 - h_1^3)}{6h}$	$b(h - h_1)$		$\frac{bh^2}{12} = Z'$ $\frac{bh^2}{24} = Z''$	$\frac{bh}{2}$
	$\frac{b^3}{6}$	b^2		$\frac{b^2+4bb_1+b_1^2}{12(b+2b_1)} h^2 = Z'$ $\frac{b^2+4bb_1+b_1^2}{12(2b+b_1)} h^2 = Z''$	$\frac{b+b_1}{2} h$
	$\frac{\sqrt{2}}{12} b^3 = 0.118\, b^3$	b^2		$\frac{bh^3-(b-b_1)h_1^3}{6h}$	$bh-(b-b_1)h_1$
	$\frac{5}{8} b^3$	$\frac{3\sqrt{3}}{2} b^2 =$ $2.598\, b^2$		$\frac{b(a'^3-f^3)+b_1(f^3+a''^3)}{3a'} = Z'$ $\frac{b(a'^3-f^3)+b_1(f^3+a''^3)}{3a''} = Z''$	$b_1h_1+bh_2$
	$\frac{5\sqrt{3}}{16} b^3$	$\frac{3\sqrt{3}}{2} b^2$		even angle $\frac{Ah}{7.2}$	$ht+(b-t)t$
	$\frac{\pi d^3}{32}$	$\frac{\pi d^2}{4}$		$Za' = \frac{(b-b_1)h^3+b_1(h-h_1)^3-6Aa'^3}{3[2h^2(b-b_1)+b_1(h-h_1)^2]}$ $A = bh - b_1h_1$	
	$\frac{\pi}{32}\left(\frac{d_1^4-d_2^4}{d_1}\right)$	$\frac{\pi}{4}(d_1^2-d_2^2)$		$Za' = \frac{2A[ba'^3+b_2a''^3-(b-b_1)f^3-(b_2-b_1)g^3]}{3[b_1h^2+(a'-f)^2(b-b_1)+(a'-f)(b_2-b_1)(2h-a'+f)]}$ $A = b(a'-f)+b_1h_1+b_2(a''-g)$	

48. To return to the formula $M = f\,Z$ again, the value of f is the same as used in (4) and M is the bending moment expressed always in terms of inch pounds, as $M = W\,l$, where W = weight in pounds, and l = length of the lever arm in inches.

Application.—Having given a beam as in Fig. 10, where W = 3600# and l = 36″ find the size of the beam that would be used in each of the following conditions: first, square section set flat; second, square section set diagonally; third, rectangular section; fourth circular section; fifth, equilateral triangular section. Assume the material to be wrought iron in every case. If f = 8,000 we have



$$(1)\begin{cases} M = f\,Z \ldots\ldots\ldots\ldots\ldots \text{Area of section} = 20.16\,\square'' \\ 129600 = 8000\,b^3 \div 6 \ldots \text{Total shear } S = 3600\,\# \\ b = 4.6'' \ldots\ldots\ldots\ldots\ldots \text{Unit shear } f\,s = 178\,\#\square'' \end{cases}$$

$$(2)\begin{cases} M = f\,Z \ldots\ldots A = 27.04\square'' \\ 129600 = 8000 \times .118\,b^3 \ldots S = 3600\# \\ b = 5.2'' \ldots\ldots\ldots\ldots fs = 133\,\#\square'' \end{cases}$$

$$(3)\begin{cases} M = f\,Z \ldots\ldots A = 17.1\square'' \\ 129600 = 8000\,b\,h^2 \div 6 \ldots\ldots S = 3600\,\# \\ \text{assume } b = 3'' \text{ then } h = 5.7'' \ldots fs = 210\#\square'' \\ \text{If } b = 2'' \text{ then } h = 7'' \text{ and the area becomes } 14\square'' \\ \quad \text{instead of } 17.1\square''. \end{cases}$$

$$(4)\begin{cases} M = f\,Z \ldots\ldots A = 23.75\square'' \\ 129600 = 8000\,\pi\,d^3 \div 32 \ldots\ldots S = 3600\,\# \\ d = 5.5'' \ldots\ldots\ldots\ldots f\,s = 151\,\#\square'' \end{cases}$$

(5) In the triangular section we have the modulus for the base $1 \div 12\,b\,h^2$ and that for the point $1 \div 24\,b\,h^2$. Taking $Z = 1 \div 24\,b\,h^2$, since this is the weakest point we have

$$(5)\begin{cases} M = f\,Z \ldots\ldots\ldots A = 28.4\,\square'' \\ 129600 = 8000\,b\,h^2 \div 24 \ldots S = 3600\,\# \\ b\,h^2 = 388.8 \ldots\ldots\ldots fs = 137\#\square''. \\ \text{but } h = b\cos 30^\circ = .866\,b. \\ b = 8.1'' \end{cases}$$

FIG. 10.

The first four beams were symmetrical in section, that is, the neutral axis passed through the center of the section, as shown in 10 of the 16 sections of the modulus table, but when the section area is not uniform as in a tee bar or triangle where the neutral axis is above or below the center of the height there is a modulus for each part. The smaller of the two should be taken as the working modulus. See Par. 90, a, Church.

The foregoing will serve to show the use of the modulus in figuring beams. It will be noticed that the rectangular beam requires the least weight of any of the beams considered. It will also be seen that the strength of the rectangular beam varies directly as the breadth, and as the square of the height. If the breadth of the beam be doubled the strength is doubled, the weight is also doubled, but if the height is doubled the weight is doubled and the strength is quadrupled. It is an advantage to have beams as light as possible for any given strength and by the above a selection can easily be made. If this were tried in comparison with the rolled steel forms such as the I beam the latter would give even better results.

49. Maximum Bending Moment:—It will be necessary at times for the designer to locate the point in a beam where the maximum bending moment takes place. In a simple beam having two supports and loaded uniformly, or at the middle, it is located at the center of the beam while in a cantilever it is at the point of support. The point of maximum moment in any case will be where the shear changes from positive to negative and vice versa.

Having a beam loaded as shown in Fig. 11, we find the reactions at A and B to be 878.4 and 721.6 pounds respectively. Now the shear diagram would be as follows:—from A to C 878.4 pounds; from C to D 878.4 — 500 = 378.4 pounds; from D to E 378.4 — 1000 = — 621.6 pounds; from E to B = — 621.6 — 100 = — 721.6 pounds. Hence according to our former statement the greatest bending moment is at D. The value of this bending moment will be $878.4 \times 95 - 500 \times 45 = 60948$ inch pounds.

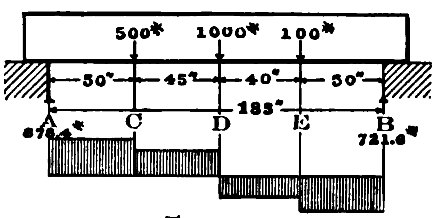


FIG. 11.

The maximum bending moment of the cantilever beams and the simple beams shown in table V, bring in only the simplest considerations. Other and more complicated loadings can be looked up in

Church's Mechanics of Engineering Pars. 239-290
Cotterills' Applied Mechanics, Pars. 28-45, 153-172
Perry's Applied Mechanics, Pars. 349-367

50. Shear:—The shearing load at any section in a beam is equal to the resultant of all the parallel forces acting on the beam on one side of the section. To illustrate, Fig. 12 gives the shear at *a b, c d*, and $e\ f = W$ for a cantilever, while for a simple beam the shear at

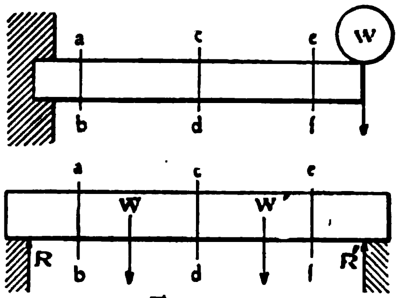


FIG. 12.

$a\ b$ is $R = W + W' - R'$

$c\ d$ is $R - W = W' - R'$

$e\ f$ is $R - (W + W') = R'$.

A beam may fail either by an excess of bending or by shearing. The former is usually the case however. If a beam is designed sufficiently strong to resist bending it generally fulfills all the requirements for shearing. Notice the values of f for flexure and shear as obtained in the application Fig. 10.

In calculating the shear of an I beam it is customary to assume that the entire shear is taken up by the web, as $b'\ h'$ Fig. 13. See Church Par. 256. Knowing this area and the total shear, the value f_s can be obtained.

51. Beams Classified:—Beams are classified as simple, cantilever, restrained and continuous.

A *simple beam* is one which rests upon two end supports.

A *cantilever beam* rests upon a support at the middle, or has one end fixed and the other end free.

A *restrained beam* or *fixed beam*, has both ends fixed.

A *continuous beam* is one which rests upon more than two supports.

For illustrations of all except continuous beams see diagrams in Table V. For continuous beams see Cases VI and XIV. Page 4, Constructor.

Wooden beams are usually rectangular in section while steel and wrought iron beams are generally as shown in Fig. 13.

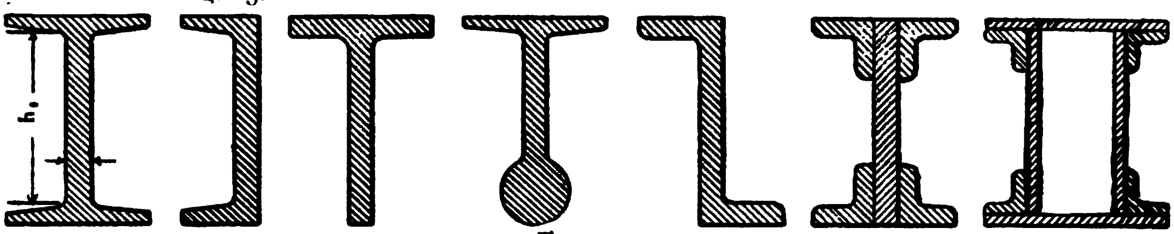


FIG. 13.

The various rolled beams are distinguished by inches in *depth of section* and *pounds per foot* of length as, a 15″ 80# beam.

52. Relative Strength of Beams:—Table V gives the relative strength of beams carrying a uniformly distributed or a concentrated load. This table was worked out from the following

Assumptions:—Total load = 2500 pounds; length of span = 20 feet; I beam from Cambria = 12″, 35#; Modulus = 38.0; area of web = 5.28□″.

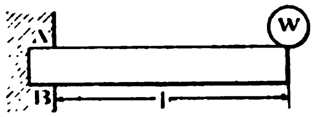
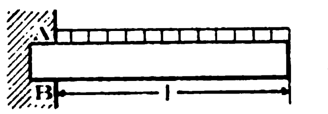
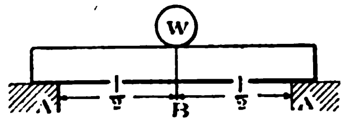
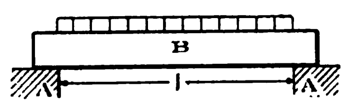
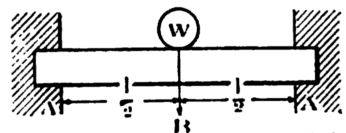
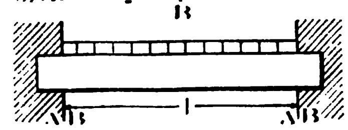
This table gives the most common forms of the applications of loads to beams. For a continuation of the table as applied to unusual sections refer to

The Constructor.................Pages 3- 4
Unwin, (Vol. I.)................Pages 54-72
Low & Bevis....................Pages 34-35

53. Deflection of Beams:—In some lines of machine design it is not necessary to determine the amount of deflection in the various members, in others, however, it would be absolutely necessary, as for example, in all forms of testing machines. The formulas given in table V for deflection may be applied in such cases.

In general, the beam member would be calculated in section for strength, first, and then the deflection formulas applied to see if the section will give sufficient rigidity. In some cases, a beam may be desired of such a size as will allow of only a certain deflection; the deflection formula would then be applied first, to obtain I, the moment of inertia of the section, from which the shape of the section would be obtained.

TABLE V.

		Max. Shear at A.	Max. Bending at B.	Deflection	Relative Strength
	Cantilever Beam Concentrated Load	$S = 2500$ $f_s = 473.4$	$M = W\,l$ $f_b = 15788$	$d = \frac{W\,l^3}{3\,E\,I}$	1.
	Cantilever Beam Uniform Load	$S = 2500$ $f_s = 473.4$	$M = \frac{W\,l}{2}$ $f_b = 7894$	$d = \frac{W\,l^3}{8\,E\,I}$	2.
	Simple Beam Concentrated Load	$S = 1250$ $f_s = 236.7$	$M = \frac{W\,l}{4}$ $f_b = 3947$	$d = \frac{W\,l^3}{48\,E\,I}$	4.
	Simple Beam Uniform Load	$S = 1250$ $f_s = 236.7$	$M = \frac{W\,l}{8}$ $f_b = 1973.5$	$d = \frac{5\,W\,l^3}{384\,E\,I}$	8.
	Fixed Beam Concentrated Load	$S = 1250$ $f_s = 236.7$	$M = \frac{W\,l}{8}$ $f_b = 1973.5.$	$d = \frac{W\,l^3}{192\,E\,I}$	8.
	Fixed Beam Uniform Load	$S = 1250$ $f_s = 236.7$	$M = \frac{W\,l}{12}$ $f_b = 1315.66$	$d = \frac{W\,l^3}{384\,E\,I}$	12.

54. Selection of Beams:—It is very convenient in specifying beams of the standard rolled sections to refer to such books as Cambria Steel, Jones & Laughlin, and Carnegie, for details of sections, section modulus, safe loads and spacing.

Illustration.—Wanted an I beam to support a ten ton moving load Fig. 14, with an allowable fibre stress of 16000 pounds per square inch.

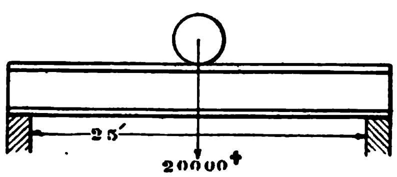


FIG. 14.

Referring to Cambria (1903) we find that all figures refer to uniformly loaded beams, however, from the preceding table it shows that a beam uniformly loaded will stand twice as much load as a beam of the same span supporting a concentrated load at the center. This fact permits the application of a uniformly loaded beam from Cambria carrying a load of 40,000 lbs. Page 82 gives a standard special B-109, 15", 80 lb. beam; also on page 84 a standard B-65, 18", 65 lb. beam which will fulfil the same conditions. Refer to page 158 for the properties of the standard beam and then to Page 6 for the shape and size of the section. Having obtained the area of the web the value f_s may be calculated. In the above case the greatest shear comes on the beam when the load is at the support then $S = 20{,}000$ lbs. From the section the area of the web of B — 65 is 7.434 square inches which gives $f_s = 2690$.

It should be noted that the fibre stress in this case was used much higher than would be ordinarily used in the designing of machine parts. Suppose some other fibre stress as 10,000, is desired as a working stress then the total load would be computed and increased by the ratio of the two fibre stresses. The beam would then be selected for this load. To illustrate: 20,000 pounds concentrated at the center or 40,000 pounds uniformly distributed will produce a fibre stress of 16,000 lbs. sq. in. in some standard section, then if we select a beam that will support 40,000 $\times$ 16 $\div$ 10 $=$ 64,000 pounds at a fibre stress of 16,000 lbs. sq. in. it will support 40,000 lbs at a fibre stress of 10,000 lbs. sq. in. Such a beam as found in Cambria, page 85 gives, No. B. 121. 20 in. Standard Special. 85 lbs.

This can be shown in another way:—

$\frac{Wl}{4} = f\,Z$; $\frac{W\,l}{4f} = Z = 150$, **which is the modulus for the above beam as shown on Page 158. Cambria.**

CHAPTER III

Notes and Formulas Continued.

55. Work:—Work is the result of force in motion and can be defined as *the overcoming of resistance along a line of motion.* Illustrations: A weight of 200 pounds is lifted 10 feet; the work done is 2000 foot-pounds. A locomotive has an average draw-bar pull in one mile of 2,000 pounds; the work done is 10,560,000 foot-pounds.

Work is independent of time.

56. Power:—Power is the *rate of doing work.* Suppose the 200 pound weight was lifted 10 feet in one second of time, the power exerted would be 2000 foot-pounds per second. The term power is usually stated as *horse-power,* the equivalent of which is 33,000 foot-pounds per minute; the above illustration would then be $2000 \times 60 \div 33{,}000 = 3.6$ H. P.

The relation between power and work, expressed as a formula is

$$\text{H. P.} = \frac{\text{P. V.}}{33{,}000} \qquad (1)$$

Where P = force in pounds and V = velocity in feet per minute.

Formula (1) is the standard horse-power formula and can be used as a basis for all power work.

57. Energy:—The kinetic energy, K. E. of a body is the *ability* of that body *to do work.* This quality expressed in terms of a formula is

$$\text{K. E.} = \frac{W v^2}{2g} \qquad (2)$$

where W is the weight of the body in pounds, v is the velocity in feet per second and $g = 32.2$.

58. Torsion:—The last of the five forces commonly met with in practice is torsion and since it relates mostly to the twisting of shafts the discussion of it was reserved until after mentioning the subject of Power.

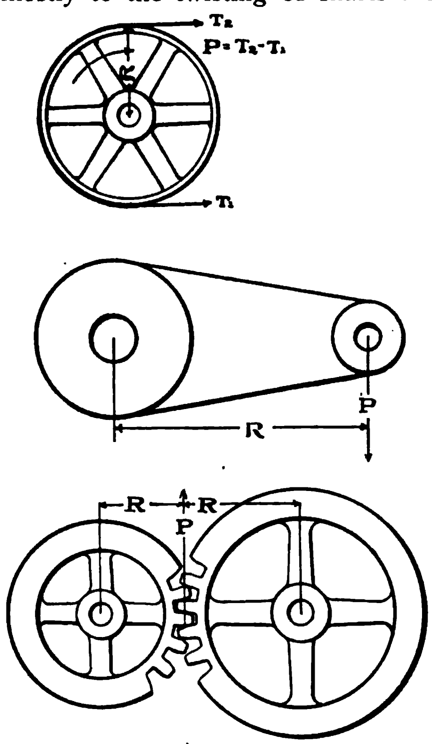


FIG. 15.

59. Twisting Moment:—Fig. 15 shows three applications of the twisting moment in machines; the pulley, the crank and the gears. In the case of the pulley let the belt while transmitting power have the two tensions T_2 and T_1 then the effective turning force of the belt is $P = T_2 - T_1$. This acts through a lever arm R giving a twisting moment P. R. The value P. R. is obtained directly from formula (1) as follows:

$$\text{H. P.} = \frac{\text{P. V.}}{33000} = \frac{2\pi \text{R. N. P.}}{33000 \times 12}$$

$$\text{P. R.} = \frac{33000 \times 12 \times \text{H. P.}}{2\pi \text{N.}}$$

N = Revolutions per minute.

It will be noticed that the moment P. R. depends only upon the H. P. and the R. P. M. The same relation shown above will be true for the crank and the gears if P. be constant.

Shafting and Shafting Supports.

60. Resistance of a Shaft to Torsion:—Any shaft subjected to a twisting moment has its fibres subjected to a shear. Imagine two planes *A B* and *A′ B′* Fig. 16, in contact and perpendicular to the center line of the shaft. If the shaft be held from turning and a force P be applied to the crank the tendency will be to turn one plane section on the other and the two lines *r*, *s*, which coincided before the application of the force, now take some position as *r s* and *r′ s′*. The strain and the stress are greatest at the outermost fibre and are zero at the center. At any intermediate point the strain and stress are proportional to the distance from the center. From this it can readily be seen that the resistance of the section (Z) is not proportional to the area but to some other factor which is found to be $\pi d^3 \div 16$. Notice that in Church Pars. 94 and 219 we find the moment of inertia of a shaft subjected to torsion to be $\pi r^4 \div 2$ and if this be divided by *r*, the distance from the center to the outermost fibre, it gives $\pi r^3 \div 2$ or $\pi d^3 \div 16$ as stated above, where d is the diameter of the shaft.

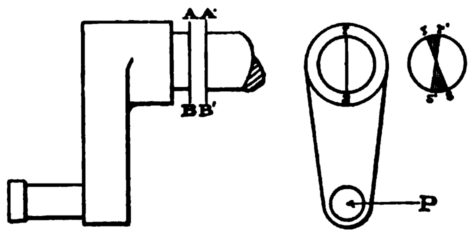


Fig. 16.

Taking f as the maximum shearing stress and substituting in the formula $M = f\ Z$ we have

$$P\,R = f\,\frac{\pi\,d^3}{16} = .19635\,d^3 f$$

$$d = 1.72\sqrt[3]{\frac{P\,R}{f}} = 1.72\sqrt[3]{\frac{T}{f}} \qquad (3)$$

For a square shaft $M = .208\ b^3 f$ where b is the side of the square.
For wrought iron shafts $f = 6000$ to 9000.
For mild steel shafts $f = 9000$ to 12000.
Formula (3) would not be used on shafts subjected to both bending and twisting.

61. Hollow Shafts:—The value of Z for a hollow shaft subject to torsion is $\pi\ (d_1^4 - d_2^4) \div 16\ d_1$ where d_1 is the outer diameter and d_2 is the inner diameter. This can then be substituted in $M = f\ Z$ as above. A very common requirement in machine design is the following:

Given a solid shaft d inches in diameter, it is desired to replace this with a hollow shaft of equal weight having d_1 inches external diameter. What is the internal diameter?; or the following: Given a solid shaft d inches diameter and capable of transmitting .... horse power under stated conditions, it is desired to replace this shaft with a light hollow shaft capable of resisting the same twisting moment and having d_1 inches external diameter: What will be the internal diameter d_2?

If the material is the same in each case we have

$$(1)\quad \frac{\text{Weight of hollow shaft}}{\text{Weight of solid shaft}} = \frac{d_1^2 - d_2^2}{d^2}$$

Which gives $d^2 = d_1^2 - d_2^2$ for equal weights.

$$(2)\quad \frac{\text{Strength of hollow shaft}}{\text{Strength of solid shaft}} = \frac{d_1^4 - d_2^4}{d_1\ d^3}$$

Which gives $d^3 = \dfrac{d_1^4 - d_2^4}{d_1}$ for equal strengths.

Application.—A two-inch shaft is transmitting 33 H. P. at 250 R. P. M. Find (1) Maximum twisting moment; (2) maximum fibre stress; (3) internal diameter d_2 of a hollow shaft 2.5 inches external diameter that is capable of resisting the same twisting moment as the solid shaft.

$$(1)\quad \text{H. P.} = \frac{2\,\pi\,R\,N\,P}{33000 \times 12}\ ;\ \text{P. R.} = 7570''\# = \text{M.}$$

$$(2)\quad f = \frac{M}{Z};\ Z = \frac{\pi\,d^3}{16};\ f = 4802\ \#\square''$$

$$(3)\quad d^3 = \frac{d_1^4 - d_2^4}{d_1};\ d_2 = 2.087''\ \text{say}\ 2''$$

If we wish a hollow shaft to be say 3 inches external diameter and of equal weight we have $d^2 = d_1^2 - d_2^2$; $d_2 = 2.25''$. A comparison of the above sizes is shown in Fig. 17.

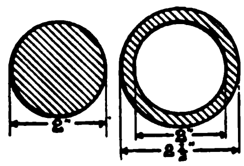
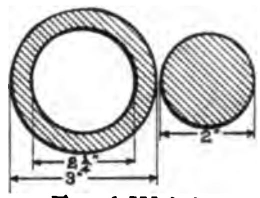


FIG. 17.



Equal Strength. Equal Weight.

62. Combined Twisting and Bending:—This combined stress is present in short shafts and in shafting and consequently the problem becomes a very important one for the engineer. Most shafting is subjected to a twisting and a bending stress simultaneously and would be classed under this head. Take for illustration a piece of shafting receiving power as in Fig. 18 with the thrust from the belt acting at the point where it will produce the greatest bending moment, i. e., midway between the bearings; we have a shearing stress in the fibres due to the twisting moment and a bending stress due to the pull of the belt. These stresses may be combined into an *equivalent shearing stress* or into an *equivalent bending stress*. To reduce to the equivalent shearing stress the method is as follows:

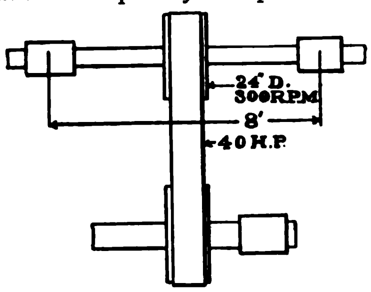


FIG. 18.

$$\text{Twisting Moment} = \text{P. R.} = \frac{\text{H P} \times 12 \times 33000}{2\,\pi\,\text{N}} = 8403\ ''\# = \text{T}_\text{m}$$

$$\text{P} = \frac{8403}{12} = 700\#$$

Let the total side thrust on the shaft due to the belt $= \text{W} = 2 \times 700 = 1400\#$.

NOTE.—Assume the ratio $\frac{T_2}{T_1} = 3$ which may be considered good condition of service; we then have from Fig. 15, $T_2 - T_1 = P$, and by combining obtain $T_2 = 3\text{ P} \div 2$. Since the total side thrust from the belt is due to $T_2 + T_1$ we have

$$T_2 + T_1 = \frac{3\text{ P}}{2} + \frac{\text{P}}{2} = 2\text{ P} = \text{W}.$$

According to this the shaft will be supporting 1400 pounds at the center as given above and will have a bending moment of

$$\frac{\text{W l}}{4} = \frac{1400 \times 12 \times 8}{4} = 33600\ ''\#$$

From Low and Bevis Par. 135 the following formula is obtained for combined twisting and bending.

$$\text{T}_\text{e} = \text{B}_\text{m} + \sqrt{\text{B}_\text{m}^2 + \text{T}_\text{m}^2} \qquad (4)$$

Where T_e = Equivalent twisting moment.
B_m = bending moment.
T_m = twisting moment.

Substituting in (4) the values of P. R. and $\frac{\text{W l}}{4}$ we have

$$\text{T}_\text{e} = 33600 + \sqrt{(33600)^2 + (8403)^2} = 68230.$$

Substituting this value of T_e into formula (3) and taking the suggested value of *f* from the same reference as 9000 we have

$$d = 1.72 \sqrt[3]{\frac{68230}{9000}} = 3.37'' \text{ say } 3\tfrac{3}{8}''$$

The conditions just worked through would be very similar to those of a *jack shaft* or of a shaft serving the same duty as a jack shaft. The shaft is very large because of the unusual bending moment. If the bending moment be reduced by making the distance between the bearings less the diameter of the shaft would become less. Suppose this be made 6 feet instead of 8 feet, our formula would then become

$$T_e = 25200 + \sqrt{(25200)^2 + (8403)^2} = 51765''\#$$

$$d = 1.72 \sqrt[3]{\frac{51765}{9000}} = 3.07'' \text{ say } 3''$$

The following empirical formulas are frequently used for mild steel shafting:—

63. High Speed Engine Shaft:—Barr, Trans. A. S. M. E., Vol. 18, Page 756.

$$d = 7.3 \sqrt[3]{\frac{\text{H. P.}}{\text{R. P. M.}}}$$

64. Dynamo Shaft:—

$$d = K_1 \sqrt[4]{\frac{W}{\text{R. P. M.}}}$$

Where d = diameter of shaft in the armature core.
W = output in watts.
K_1 = constant depending on output as per table.

K W	1	5	10	50	100	200	500	1000	2000
K_1	1	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8

65. Counter Shafts subjected to a heavy bending load:—

$$d = \sqrt[3]{\frac{125 \text{ H. P.}}{\text{R. P. M.}}}$$

66. Ordinary Line Shafting subjected to both bending and twisting:—

$$d = \sqrt[3]{\frac{90 \text{ H. P.}}{\text{R. P. M.}}}$$

67. Transmission Shafting in which the only bending stress is that due to its own weight:—

$$d = \sqrt[3]{\frac{62.5 \text{ H. P.}}{\text{R. P. M.}}}$$

68. Classification of Shafts and Shafting:—

Prime Movers: Engine Shaft. Dynamo Shaft. Turbine Shaft.

Transmission Lines: Jack Shaft. Lines of Shafting. Counter Shaft.

Machine Shafts.

69. Materials Used in Shafting:—Wrought iron and mild steel. Steel shafting is more rigid, can be made more cheaply and is replacing wrought iron shafting to a large degree.

70. Process of Manufacture:—
Hot Rolled.—Dark rough surface.
Cold Rolled.—Bright smooth surtace.

71. **Preparation for the Market**:—

Black Shafting.—Turned only at journals and couplings; not much used.

Bright Shafting.—Turned wrought iron or steel; cold rolled wrought iron or steel.

Turned shafting is made from standard hot-rolled rods by turning off $\frac{1}{16}$ inch in diameter. The following table gives the standard sizes prepared for the market:

Size of Rod in inches...............	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2¼	2½	3
Size of Shafting in inches............	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{9}{16}$	$1\frac{11}{16}$	$1\frac{13}{16}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{7}{16}$	$2\frac{15}{16}$

Cold rolled shafting is made by drawing it through a die the exact diameter of the finished shafting. Cold rolled shafting is about 25% better to resist twisting and bending than turned shafting of the same diameter and the same material.

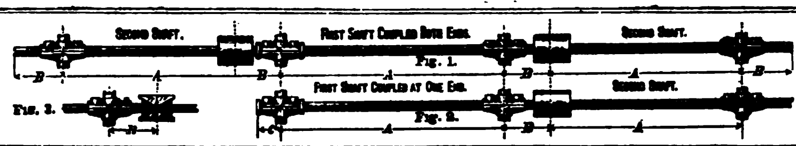
72. **Length of Shafts**:—Shafting is manufactured in standard lengths but it can be ordered cut to almost any length up to twenty-five feet.

It is always wise in ordering shafting to specify the exact diameter, length, kind of material and finish; as *20 feet of 2 inch cold rolled steel shafting* or *20 feet of turned steel shafting $1\frac{15}{16}$ inches actual diameter.* The reason for this is because the turned shafting is usually rated the same diameter as the rough rod from which it was turned; thus a piece of 2 inch turned shafting is $1\frac{15}{16}$ inches actual diameter.

The following table taken from Wm. Sellers & Co.'s catalog may be valuable in planning shafting lines:

TABLE VI.—LAYING OUT SHAFTS.

In Coupling Shafts of different sizes, either use a Coupling to suit the larger Shaft, with one side bored for the smaller one, or reduce the end of the large shaft in diameter, and use a small Coupling.



Make Bearings at equal distances from each other, when practicable, and always put Two Bearings on the First, which is the Collared Shaft. See Figs. 1 and 2.

DISTANCE FROM CENTRE OF BEARING TO END OF SHAFT FOR COUPLING. See B, Figures 1, 2, and 3.

	Length of Collar	Size of 1st Shaft	1 7/16"	1 11/16"	1 15/16"	2 3/16"	2 7/16"	2 11/16"	2 15/16"	3 3/16"	3 7/16"	3 15/16"	4 7/16"	4 15/16"	5 7/16"	5 15/16"	6 7/16"	6 15/16"	7 7/16"	7 15/16"	Size of Shaft	Length of Standard Bearing	Double Cone-Vise Coupling. Length	Double Cone-Vise Coupling. Diam.	Size of Keyseats in Shaft and Coupling. Width	Size of Keyseats in Shaft and Coupling. Depth
For Loose Collars	3 13/16"	1 7/16"	8½"																		1 7/16"	6"	5½"	4½"	5/16"	5/16"
	4 5/16"	1 11/16"	9"	9½"																	1 11/16"	7"	6½"	4½"	5/16"	5/16"
	5⅛"	1 15/16"	9½"	10½"	11"																1 15/16"	8"	7½	5½"	7/16"	¼"
	5 11/16"	2 3/16"		11"	11½"	12"															2 3/16"	9"	8½"	5½"	7/16"	¼"
	6¼"	2 7/16"		11½"	12"	12½"	13"														2 7/16"	10"	9½"	6 3/16"	9/16"	5/16"
	6 13/16"	2 11/16"		12"	12½"	13"	13½"	14"													2 11/16"	11"	10"	7"	11/16"	⅜"
	7 7/16"	2 15/16"			13"	13½"	14"	14½"	15"												2 15/16"	12"	11"	7½"	11/16"	⅜"
	8⅛"	3 3/16"			13½"	14"	14½"	15"	15½"	16½"											3 3/16"	13"	12"	8½"	13/16"	⅜"
	8¾"	3 7/16"			14"	14½"	15"	15½"	16"	16"	16½"										3 7/16"	14"	13"	9 3/16"	13/16"	⅜"
	9½"	3 15/16"			15"	15½"	16"	16½"	17"	17"	17½"	18"									3 15/16"	16"	14½"	10"	15/16"	7/16"
	10¼"	4 7/16"				16½"	17"	17½"	18"	18"	18½"	19½"	20"								4 7/16"	18"	16"	11⅝"	15/16"	7/16"
	11⅛"	4 15/16"					18½"	18½"	19"	19½"	20"	20½"	21½"	22"							4 15/16"	20"	18"	12 3/16"	15/16"	7/16"
For Fast Collars	12 5/16"	5 7/16"						20"	20½"	21"	21"	22"	22½"	23½"	24"						5 7/16"	22"	19½"	13½"	1 1/16"	½"
	13⅛"	5 15/16"						21½"	21½"	22"	22½"	23"	24"	24½"	25"	26"					5 15/16"	24"	21"	14½"	1 1/16"	½"
	14⅛"	6 7/16"							23"	23½"	24"	24½"	25"	26"	26½"	27½"	28½"				6 7/16"	26"	25"	17½"	1 1/16"	½"
	15¼"	6 15/16"								25"	25"	26"	26½"	27½"	28"	29"	30"	30½"			6 15/16"	28"	25"	17⅝"	1 3/16"	9/16"
	16¼"	7 7/16"									26½"	27"	28"	28½"	29½"	30"	31"	31½"	32"		7 7/16"	30"	28½"	19 5/16"	1 3/16"	9/16"
	17⅛"	7 15/16"										28½"	29½"	30"	31"	31½"	32½"	33"	34"	34½"	7 15/16"	32"	28⅝"	19 5/16"	1 3/16"	9/16"

USE OF TABLE.

Look for size of first Shaft in left-hand column, under the head of size of first Shaft, and in the top line of table, marked size of second Shaft, find the size of the Shaft to be coupled to it. The intersection gives the length B, this added to the length A, or distance from centre to centre of Bearing, and in cases similar to Fig. 2, to the length C, gives the length of the first Shaft, thus: as in Fig. 1. B + A + B = Length. Fig. 2. C + A + B = Length.

22 22

Wherever it is possible to do so, pulleys should be set next to the journals to relieve the shaft from the bending due to the pull of the belt.

73. **Speed of Shafting**:—The best speed for shafting lines is 300 R. P. M. This may be reduced for iron working machinery to 200 R. P. M. and may be increased for wood working machinery to 400 R. P. M.

74. **Speed of Counter Shafts**:—

Iron Working Machines 80 to 150 R. P. M.

Wood Working Machines 200 to 1000 R. R. M.

75. Hangers and Shaft Supports:—The following diagrams show some of the different methods of supporting shafts. Data referring to these various forms will be found on pages 25, 26 and 27.

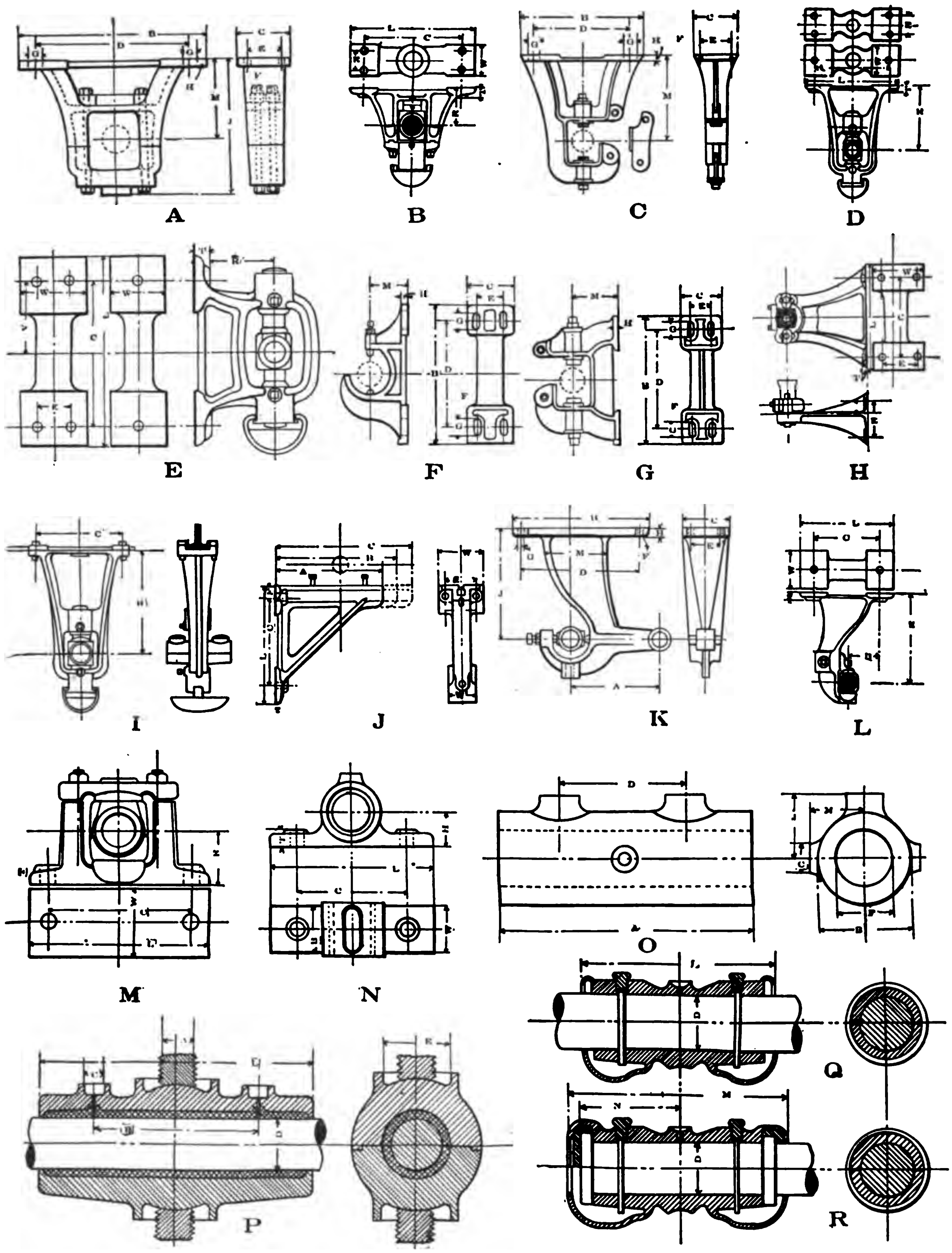


FIG. 19.

TABLES VII

A to R.

HEAD-SHAFT HANGERS.—A

Size of Shaft in inches	B Inch.	C Inch.	D Inch.	E Inch.	F 4 Bolts. Diam. Inch.	G Inch.	H Inch.	J Inch.	M Inch.
2¼—3	25	6	21½	3⅛	⅞	2	1⅛	18½	12
3½—4	28	8	24	5	1	2¼	1¼	19¾	12
4¼—5	31	9	26	5¾	1⅛	2½	1⅜	21	12
5¼—6	34	10	29	6⅜	1¼	2¾	1½	22	12
6¼—7	36½	11	31½	7¼	1⅜	3	1¾	23¼	12
7¼—8	38	12	32½	7½	1½	3	2	24½	12

HEAD-SHAFT HANGERS—B

Size of Shaft	R Drop	L Length of Foot	W Width of Foot	T Thick-ness of Foot	C Centre to Centre of Bolts	E Centre to Centre of Bolts	Numb'r and Size of Bolts
3 15/16"	11"	34"	7⅛"	1¼"	26"	4¼"	4—1"
4 7/16"	11"	35⅝"	8 5/16"	1 5/16"	27⅝"	5"	4—1"
4 15/16"	11"	37¼"	9⅛"	1⅜"	28¾"	5½"	4—1"
5 15/16"	11"	40½"	10¼"	2⅛"	31½"	6"	4—1¼"

BALL AND SOCKET DROP HANGER FRAMES—C

Size of Shaft in inches	M	B	C	D	E	G	H	No. and Diam. of Bolts at F
2 7/16 to 2½	10	17	5	13¾	......	1½	¾	2—⅝
	12	18½	5⅛	14½	......	1⅜	¾	2—⅝
	14	18⅜	5⅝	14⅛	2⅝	1⅞	¾	4—¾
	16	19⅞	6	15¾	3⅛	1⅞	¾	4—¾
	18	21⅝	6⅜	17⅜	3¼	1⅞	⅞	4—¾
	20	23⅛	6¾	18½	3¼	1⅞	⅞	4—¾
	24	26⅞	7⅞	21¼	4¼	2⅛	1	4—¾
	30	26⅛	8⅝	20⅞	4⅝	2⅜	1	4—¾
	36	30¼	8½	24¾	4⅝	2⅛	1	4—⅞
2 15/16 to 3	10	16¼	5½	12⅛	2¾	1⅜	¾	4—⅝
	12	17⅛	5⅞	13¼	3	1⅛	⅞	4—⅝
	14	17⅞	6	13⅝	3¼	1⅜	⅞	4—¾
	16	19⅝	6⅛	15¾	3½	1⅞	⅞	4—¾
	18	21⅝	6½	17½	3½	1⅞	1	4—¾
	20	24⅛	7⅝	19¼	4⅛	2⅛	1	4—⅞
	24	28⅞	9	23⅜	5	2⅛	1⅛	4—⅞
	30	29¾	9⅛	23⅛	4⅝	2⅝	1⅛	4—⅞
	36	32¼	9⅛	26	4⅛	2½	1⅛	4—1
3 7/16 to 3½	12	18	6⅝	14¼	3⅝	1¾	1	4—⅞
	14	20	6⅛	15⅝	3⅜	1⅞	1	4—⅞
	16	20⅛	6⅛	16⅛	3⅛	1¾	1	4—⅞
	18	23½	8⅛	18⅝	4¾	2¼	1⅛	4—⅞
	20	27	8⅝	21¾	5	2½	1⅛	4—⅞
	24	31¾	10	25½	6	2¾	1⅛	4—⅞
	30	30⅜	8¾	25	5⅛	2⅜	1⅛	4—1
	36	35⅞	9¼	30¾	5⅞	2⅜	1⅛	4—1⅛

BALL AND SOCKET LINE HANGERS—D and I

R Drop Inches	L Length of Foot Inches	W Width of Foot Inches	T Thickn'ss of Foot Inches	C Centre to Centre of Bolts Inches	E Centre to Centre of Bolts Inches	No. and Size of Bolts No. In.
1 7/16" HANGER						
8	13⅛	5	¾	10⅛	....	2—⅝
10	13⅜	5½	¾	10⅜	....	2—⅝
11	14⅛	5¾	¾	11⅛	....	2—⅝
13	14⅝	6	¾	11⅝	....	2—⅝
16	17⅜	6⅛	¾	14⅜	....	2—⅝
18	18¾	6½	¾	15⅛	3¼	4—½
20	20	6½	¾	16¾	3½	4—½
1 15/16" HANGER						
8	14⅞	5½	⅞	10⅞	....	2—¾
10	15¾	6	⅞	11⅜	....	2—¾
11	16¾	6¼	⅞	12¾	....	2—¾
13	17⅝	6½	⅞	13⅝	....	2—¾
16	19¾	7	⅞	15¾	....	2—¾
18	21	7¼	⅞	17	4	4—⅝
20	22¼	7½	⅞	18¼	4½	4—⅝
25	25⅛	9	⅞	21⅛	5⅛	4—⅝
30	27½	10	⅞	23½	6	4—⅝
2 7/16" HANGER						
8	17	6	1	12	....	2—⅞
10	18¼	6⅜	1	13¼	....	2—⅞
11	18¾	6¾	1	13¾	....	2—⅞
13	20¼	7	1	15¼	....	2—⅞
16	22¼	7⅛	1	16⅞	....	2—⅞
18	24	7½	1	18¾	4	4—¾
20	25	8	1	20	4¾	4—¾
25	27¼	9½	1	22¼	5¾	4—¾
30	30⅝	10⅛	1	25⅝	6¼	4—¾
2 15/16" HANGER						
8	19	6½	1¼	13	....	2—1
10	20¼	6¾	1¼	14¼	....	2—1
11	21 1/16	7¼	1¼	15 1/16	....	2—1
13	22	7½	1¼	16	....	2—1
16	24¼	8	1¼	18¼	....	2—1
18	25¼	8¼	1¼	19¼	4¾	4—⅞
20	26½	8½	1¼	20½	5	4—⅞
25	29⅜	10	1¼	23⅜	6	4—⅞
30	32⅜	11	1¼	26⅜	6½	4—⅞
36	36¼	12	1¼	30¼	7¼	4—⅞
3 7/16" HANGER						
10	21½	7½	1⅜	14⅜	....	2—1
11	23 3/16	7¾	1⅜	16 3/16	....	2—1
13	25¼	8	1⅜	17¼	....	2—1
16	26⅛	8⅛	1⅜	19⅛	....	2—1
18	27	8¾	1⅜	20	5	4—⅞
20	28¾	9	1⅜	21¾	5¼	4—⅞
25	31⅝	10½	1⅜	24⅜	6⅛	4—⅞
30	34⅜	11⅛	1⅜	27⅜	6¾	4—⅞
36	37⅞	12⅛	1⅜	30⅞	7¼	4—⅞
3 15/16" HANGER						
10	25 5/16	8¼	1½	17 5/16	....	2—1⅛
11	25 5/16	8¼	1½	17 5/16	....	2—1⅛
13	26 5/16	8½	1½	18 5/16	....	2—1⅛
16	28⅜	9	1½	20⅜	....	2—1⅛
18	29⅝	9¼	1½	21⅝	5	4—1
20	30 13/16	9½	1½	22 13/16	6	4—1
25	33⅞	11	1½	25⅞	6½	4—1
30	37⅛	12	1½	29⅛	7	4—1
36	40⅝	13	1½	32⅝	7½	4—1
4 15/16" HANGER						
11	29¾	9¼	1⅞	19½	5¼	4—1⅛
13	30	10	1⅞	20	5¾	4—1⅛
16	32	10½	1⅞	22	6¼	4—1⅛
18	34½	10½	1⅞	24½	6¼	4—1⅛
20	35½	10½	1⅞	25⅛	6¾	4—1⅛
25	38½	12	1⅞	28½	7¾	4—1⅛

DIMENSIONS OF BALL-AND-SOCKET POST HANGERS—E

For Ring Self-Oiling Boxes

Size of Shaft	L Length of Foot	W Width of Foot	T Thickness of Foot	C Centre to Centre of Bolt Holes	E Centre to Centre of Bolt Holes Horizontal	V Upper Bolt Hole above Centre of Shaft	R Foot to Centre of Shaft	Diameter of Bolts
Inch's	Inch's	Inch's	Inches	Inches	Incher	Inches	Inches	Inches
1 7/16	12	3 3/4	11/16	9	....	4 3/4	6	5/8
1 11/16	13 3/8	3 7/8	3/4	9 7/8	....	5 1/8	6	5/8
1 15/16	15	4	7/8	11	....	5 3/4	6	3/4
2 3/16	16 1/2	4 1/2	7/8	12	....	6 3/8	6	3/4
2 7/16	18	5	1	13	....	6 3/4	6	7/8
2 11/16	19	5 1/2	1 1/8	13 1/2	....	7 1/4	6	7/8
2 15/16	20	6	1 1/4	14	....	7 7/8	6	1
3 3/16	22	6 1/2	1 5/16	15 1/2	....	8 3/8	6	1
3 7/16	20 1/4	9 1/2	1 3/8	14	5 1/2	7 3/4	6	7/8
3 15/16	22 1/8	9 3/4	1 1/2	15 1/8	6	8 3/8	6	1
4 7/16	24 1/8	10	1 1/2	16 5/8	6 3/4	10	7 1/2	1 1/8
4 15/16	25 3/4	10 1/2	1 5/8	17 3/4	7 1/2	10 3/4	7 1/2	1 1/4

FOR STANDARD BOXES

1 7/16	12	3 3/4	11/16	9	....	4 3/4	6	5/8
1 11/16	13 3/8	3 7/8	3/4	9 7/8	....	5 1/8	6	5/8
1 15/16	15	4	7/8	11	....	5 3/4	6	3/4
2 3/16	16 1/2	4 1/2	7/8	12	... .	6 3/8	6	3/4
2 7/16	18	5	1	13	....	6 3/4	6	7/8
2 11/16	19	5 1/2	1 1/8	13 1/2	....	7 1/4	6	7/8
2 15/16	20	6	1 1/4	14	....	7 7/8	6	1
3 3/16	22	6 1/2	1 5/16	15 1/2	....	8 3/8	6	1
3 7/16	20 1/4	9 1/2	1 3/8	14	5 1/2	7 3/4	6	7/8
3 15/16	22 1/8	9 3/4	1 1/2	15 1/8	6	8 3/8	6	1
4 7/16	24 1/8	10	1 1/2	16 5/8	6 3/4	10	6	1 1/8
4 15/16	25 3/4	10 1/2	1 5/8	17 3/4	7 1/2	10 3/4	6	1 1/4

WALL BRACKET HANGERS—F

Size of Shaft in inches	B	C	D	E	No. and Size of Bolts. F	G	H	M
1 3/16 to 1 9/16	13 3/4	4	11 1/2	.....	2—5/8	1 3/8	3/4	2 1/4
1 5/8 to 1 15/16	14 1/4	4 3/8	11 5/8	......	2—5/8	1 1/4	7/8	2 1/2
1 7/8 to 2 1/16	14 3/4	4 1/2	12	......	2—5/8	1 1/2	7/8	3
2 1/8 to 2 5/16	15 1/2	4 7/8	12 5/8	......	2—5/8	1 5/8	1	3 5/8
2 3/8 to 2 5/8	16 3/8	5	13 1/4	......	2—5/8	1 3/4	1	3 3/4
2 3/4 to 2 7/8	17 1/2	5 1/2	14 1/2	.....	2—3/4	1 7/8	1	3 7/8
2 15/16 and 3	17 7/8	5 3/4	14	.....	2—7/8	2	1	4 5/8
3 3/16 and 3 1/4	17 7/8	6	13 1/2	3 1/2	4—3/4	2 3/8	1	4 3/4
3 7/16 and 3 1/2	19 1/2	6 5/8	15	3 3/4	4—7/8	2 1/2	1	5 3/8
3 3/4 to 3 7/8	20 1/8	6 5/8	15 3/4	3 7/8	4—7/8	2 5/8	1	5 1/2
3 15/16 and 4	21	7	16 1/8	4	4—7/8	2 3/4	1	5 7/8

WALL BRACKET HANGERS—G

Size of Shaft, inches	B	C	D	E	No. and Size of Bolts. F	G	H	M
1 to 1 1/4	16	4 1/4	13 1/8	.....	2—5/8	1 1/4	7/8	5 7/8
1 15/16 and 2	18 1/2	4 1/2	15 1/2	.....	2—5/8	1 3/8	7/8	5 7/8
2 3/16 to 2 1/2	19 1/2	5 1/4	15 5/8	2 3/8	4—5/8	1 7/8	7/8	6 1/4
2 15/16 to 3	20 3/4	6 3/8	16 3/4	3 1/4	4—3/4	1 3/4	7/8	6 1/2
3 3/16 to 3 1/2	21 1/2	7 3/4	17 1/4	4 1/4	4—3/4	2	1	8
3 15/16 to 4	21 5/8	7 1/2	17 1/8	3 3/4	4—7/8	2 1/8	1	8 1/8
4 3/16 to 4 1/2	25 3/8	9	20 5/8	5 1/8	4—7/8	2	1 1/8	8 1/8

DIMENSIONS OF WALL HANGER BRACKETS—H

No. of Bracket	Size of Shaft	L	W	T	C	E	F	G	Size of Bolts	Projection from Wall
	Inch.	In.	In.	In.	In.	In.	In.	In.	In.	Inches
20	1 7/16 1 11/16 1 15/16	18	9	7/8	13 1/2	6	2	1 13/32 1 23/32 1 29/32	5/8 5/8 5/8	a=13 1/2 to 15 1/2 b=15 1/2 to 17 1/2 c=17 1/2 to 19 1/2
21	1 7/16 1 11/16 1 15/16	22	10	7/8	17 1/2	7	2 3/8	1 13/32 1 23/32 1 29/32	5/8 5/8 5/8	d=19 1/2 to 21 1/2 e=21 1/2 to 23 1/2 f=23 1/2 to 25 1/2
22	1 7/16 1 11/16 1 15/16	25	11	7/8	20 1/2	8	2 1/2	1 13/32 1 23/32 1 29/32	5/8 5/8 5/8	g=25 1/2 to 27 h=27 to 29 i=29 to 31
23	2 3/16 2 7/16 2 11/16	22	9 1/4	1 1/8	17	7	2	2 1/32 2 9/32 2 11/32	3/4 3/4 3/4	a=13 1/2 to 15 1/2 b=15 1/2 to 17 1/2 c=17 1/2 to 19 1/2
24	2 3/16 2 7/16 2 11/16	25	10 3/4	1 1/8	20	7 3/4	2 3/8	2 1/32 2 9/32 2 11/32	3/4 3/4 3/4	d=19 1/2 to 21 1/2 e=21 1/2 to 23 1/2 f=23 1/2 to 25 1/2
25	2 3/16 2 7/16 2 11/16	28	11 3/4	1 1/8	23	8 1/2	2 7/8	2 1/32 2 9/32 2 11/32	3/4 3/4 3/4	g=25 1/2 to 27 h=27 to 29 i=29 to 31
26	2 15/16 3 3/16 3 7/16	25	10 1/2	1 1/4	19	7 1/8	2 1/2	2 13/32 2 3/4 3 1/16	7/8 7/8 7/8	a=13 1/2 to 15 1/2 b=15 1/2 to 17 1/2 c=17 1/2 to 19 1/2

DIMENSIONS OF WALL BRACKETS FOR PILLOW BLOCKS—J

Number of Bracket	L Length of Base	W Width of Upper Foot	W' Width of Lower Foot	T Thickness of Feet	F Distance from Top of Bracket to Upper Bolt Holes	E Distance between Upper Bolt Holes	C Distance between Upper and Lower Bolt Holes	Projection from Wall: a Least	b Mean.	c Greatest	Size of Bolts
	Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches
1	14	6	3	1 1/4	1 1/2	4	11	13	15	17	5/8
2	15	6	3	1 1/4	1 1/2	4	12	19	21	23	5/8
3	18	6 1/2	3	1 1/4	1 1/2	4 1/2	15	25	27	29	5/8
4	20 1/4	7	4	1 1/4	1 1/2	5	17 1/8	31	33	35 1/2	5/8
5	16 3/4	6 3/4	4	1 1/2	1 3/4	4 3/4	13	15 1/2	18 1/2	21 1/2	3/4
6	19 3/4	7 1/4	4	1 1/2	1 3/4	5	16	23 1/2	26 1/2	29 1/2	3/4
7	25	7 3/4	5	1 1/2	1 3/4	5 1/4	21 1/4	31 1/2	34 1/2	37	3/4
8	22	8	5 1/4	1 3/4	2	5 1/2	17 1/4	19 3/4	24 1/4	28 3/4	7/8
9	28 1/4	9 1/2	5 1/2	1 3/4	2	7	23 1/4	31 1/4	34 3/4	38	7/8
10	25 1/4	9 1/2	7	2	2 1/2	6 1/2	19 1/4	23	27	30 1/4	1
11	31 1/2	11	7	2	2 1/2	7 1/2	25 1/2	33	36 1/2	40 1/2	1
12	28 1/2	11	7	2 1/4	3	7 1/2	21	26	30	34	1 1/8
13	36	13	9	2 1/4	3	9	28 1/2	36 1/2	39 1/2	42 1/2	1 1/8

COUNTER HANGERS—K

Size of Shaft in Inches	B Inch.	C Inch.	D Inch.	E Inch.	F 4 Bolts Diam. Inch.	G Inch.	H Inch.	J Inch.	M Inch.
1½—1¾	10¼	4½	7¾	3⅛	½	1⅜	1	7¼	3½
2	11	5½	8⅜	4⅛	⅝	1⅜	1⅛	8¼	4
2¼—2½	12⅛	6½	9¾	4¾	⅝	1⅞	1¼	9⅝	4½
2¾—3	14¼	7¼	10¾	5½	¾	2	1⅜	11⅛	5⅜
3¼—3½	16¼	7¾	12¼	6	¾	2½	1½	12½	6
3¾—4	19¾	9	14½	6¾	⅞	3⅛	1 9/16	13¾	6½

DIMENSIONS OF COUNTER HANGERS—L
With or Without Belt Shifter Arms.

Size of Shaft Inch.	R Drop Inch.	L Length of Foot Inch.	W Wid'h of Foot Inch.	T Thickness of Foot Inches	C Centre to Centre of Bolts Inches	E Centre to Centre of Bolts Inches	B Centre of Box to 1st Bolt Inches	No. and Size of Bolts Inches
1 7/16	8	10⅝	4⅜	¾	6⅞	....	2¾	2-⅝
	10	11¼	5	¾	7¾	....	3 13/16	2—⅝
	13	12⅜	5⅛	¾	8⅝	....	3 15/16	2—⅝
	16	14¼	5⅝	¾	11⅝	....	4 13/16	2—⅝
	20	17⅛	6⅛	¾	13⅝	3½	6⅜	4—½
	25	19½	6¾	¾	16¼	3¾	8 1/16	4—½
1 9/16	13	12⅞	5¾	¾	9¾	....	3⅞	2-⅝
	16	14¾	5⅞	¾	11¾	....	4⅞	2—⅝
1 11/16	8	10⅝	4⅞	¾	7⅛	....	1¼	2—⅝
	10	11⅛	5⅛	¾	8	....	3⅛	2—⅝
	13	13½	5¾	¾	10	....	4⅞	2—⅝
	16	15¼	5⅞	¾	11¾	....	4 9/16	2-⅝
	20	17⅞	6⅝	¾	14¼	3¾	7	4—½
	25	20⅜	7	¾	17⅜	4	9 9/16	4—⅛
1 15/16	13	13⅜	5⅞	⅞	9⅛	....	3 11/16	2—¾
	16	15¾	6¼	⅞	11⅞	....	4⅞	2—¾
	20	18⅛	7	⅞	14⅛	3¾	7¼	4—⅝
	25	21	7¼	⅞	17	4	7 3/16	4—⅝

DIMENSIONS OF PILLOW BLOCKS—M
With Standard Boxes

Size of Shaft Inches	L Length of Foot Inches	W Width of Foot Inches	T Thickness of Foot Inches	C Centre to Centre of Bolt Holes Inches	R Base to Centre of Shaft Inches	Number and Diameter of Bolts Inches
1 3/16	6½	2½	11/16	5	1⅞	2— ½
1 7/16	7¼	2¾	¾	5¾	2	2— ½
1 9/16	7⅞	3	¾	6⅞	2⅛	2— ½
1 11/16	8⅜	3⅜	15/16	7⅛	2½	2— ½
1 15/16	10¼	3¾	1⅛	8	2¾	2— ⅝
2 3/16	10⅞	4¼	1 3/16	8½	3⅛	2— ¾
2 7/16	11½	4⅝	1 5/16	9	3⅜	2— ¾
2 11/16	12⅞	5	1 7/16	10	3¾	2— ⅞
2 15/16	13½	5½	1 9/16	10¾	4	2— ⅞
3 3/16	14	5⅞	1⅝	11¼	4¼	2— ⅞
3 7/16	16	6¾	1 13/16	12½	4⅞	2—1
3 15/16	18	7⅛	2	14	5½	2—1
4 7/16	20¾	8⅜	2⅛	16¾	6⅛	2—1⅛
4 15/16	21⅞	9⅛	2½	17¼	6 11/16	2—1¼
5 7/16	24	10	2⅝	19½	7⅜	2—1½
5 15/16	25¼	10¾	2⅞	20½	8	2—1½

SOLID PILLOW BLOCK—N

	Inches	Inches	Inches	Inches	Inch.	Inch.
Size of Bore............	1 15/16	2 3/16	2 7/16	2 15/16	3 7/16	3 15/16
B. Length of bearing..	6	6¾	7½	9	10½	12
L. Length of foot......	8½	9¾	10¾	11¾	13½	14½
W. Width of foot.......	3½	4	4½	5½	6½	7½
T. Thickness of foot...	1	1⅛	1¼	1½	1¾	2
H. Height to centre....	2	2¼	2½	3	3½	4
C. Between bolt centres	6¼	8	8¼	9	11	12
Size of bolts	⅝	¾	¾	⅞	1	1

SOLID JOURNAL BOX—O

F Size of Shaft in Inches	A Inches	B Inches	C Inches	D Inches
15/16	4	7¼	2½	5¼
1 3/16	4⅞	7¼	3	5⅝
1 7/16	4⅞	9	3	6⅞
1 11/16	5⅝	10	3⅜	7½
1 15/16	6⅛	10½	3½	7⅞
2 3/16	6⅞	11	3¾	8
2 7/16	7⅝	11½	4	8¾
2 11/16	8¼	12½	4¼	9½
2 15/16	8¾	12½	4¾	9
3 3/16	9⅞	13	5	9½
3 7/16	11	13½	5½	10
3 11/16	11⅜	15	5⅝	11
3 15/16	12¼	15⅞	6	11¼

SPLIT JOURNAL BOX—P

D Size of Shaft in Inches	A	B	C	L	E
15/16	¾	3⅜	½	5½	2¼
1 3/16	⅞	4	⅝	6¼	2½
1 5/16	15/16	4¼	⅝	6¾	2⅝
1 7/16	1 1/16	4⅝	⅝	7⅛	2⅞
1 11/16	1⅛	5¼	11/16	7½	3
1 15/16	1⅜	5⅞	11/16	8⅜	3¼
2 3/16	1⅜	6⅛	¾	9	3⅜
2 7/16	1 9/16	7	¾	10⅛	3¾
2 11/16	1 9/16	7⅝	⅞	10½	3¾
2 15/16	1 11/16	8¼	⅞	10⅞	4 1/16
3 7/16	1 13/16	9½	15/16	12	4 9/16
3 15/16	2 1/16	10¾	1 1/16	13	5
4 7/16	2⅛	11⅞	1 3/16	13¾	5 5/16
4 15/16	2½	13⅛	1 3/16	15⅝	6
5 7/16	2¾	14¾	1 3/16	16⅞	6½
5 15/16	3	16	1⅜	18⅜	7
6 7/16	3¼	17½	1 9/16	19⅞	7½

SELF OILING BEARINGS—QR
With or Without Collars

Size of Shaft	3 Diam. Box		4 Diam. Box	
	M With Collars	L Without Collars	M With Collars	L Without Collars
Inches	Inches	Inches	Inches	Inches
1 7/16			9⅛	7¼
1 11/16	8½	6 7/16	10¼	8 3/16
1 15/16	9⅜	7⅜	11⅜	9⅜
2 3/16	10 7/16	8 3/16	12 11/16	10 7/16
2 7/16	11⅜	9	13⅞	11½
2 11/16	12 9/16	9 13/16	15 1/16	12 9/16
2 15/16	13⅜	10⅞	16⅜	13⅞
3 3/16	14½	11⅝	17¾	14⅞
3 7/16	15⅝	12½	19⅛	16
3 15/16	17¾	14¼	21¾	18¼
4 7/16	19⅜	15⅞	23⅞	20⅜
4 15/16	20⅞	17⅜	24⅞	22⅜
5 7/16	23	19	28½	25½
5 15/16	24⅛	20¼	30⅛	26¼
6 7/16	28	24	35	31
6 15/16	28	24	35	31

76. Journals:—The average distance from center to center of journals, on lines of wrought iron or steel shafting carrying belts and pulleys, varies from 8 feet on a 2 inch line to 16 feet on a 5 inch line. The *maximum allowable distance* is

2″	12′
3″	15′
4″	18′
5″	22′

77. Length of Journal:—The length of the journal is taken from 3d to 4d.

78. Journal Box:—Journal boxes are made both solid and split. Arrangements for oiling are made as follows: first, by gravity feed oil cups; second, through a wick which dips down into an oil well in the box and laps over the top of the shaft; third, by a ring or chain which spans the shaft and has its lower part emersed in the oil in the well. When the shaft revolves this ring or chain moves with it thus carrying oil to the top of the shaft. The second and third are called *self oiling boxes*. Counter shafts usually have solid boxes with gravity feed oil cups. Line shafts usually have split boxes arranged for either wick, chain or ring oiling.

BELT TRANSMISSION.

79. Belt Transmission Materials:—Belting is classified as to material, as leather, canvas and rubber. Leather is the standard belting for shop and power plant service. It should be kept dry and should not be used in any temperature above 110° F. Leather belting is made of strips of leather, which have been cut from the hides and joined together by glue or cement, and rivets. To make a thick belt, one or more of these layers are glued together hence the classification single, double, etc. For ultimate strength of leather belting see Par. 81.

Canvas belting is made from a number of layers of canvas stitched together and finally *sized*. The thickness agrees fairly well with that of leather belt. It can be used in damp places if necessary. Canvas belting has an ultimate tensile strength of about 5800 pounds per square inch. It is used largely on conveying machinery.

Rubber belting gives better contact to the pulley than leather or canvas, and is not affected by moisture. It is more nearly uniform in width and thickness, will stand wide variation in heat and cold, and has a tensile strength of about 3500 pounds per square inch.

The under line of a horizontal belt should be the working line, in which case the sag of the upper line increases the arc of contact.

Single thickness belts should not be used much over 12 inches in width.

The *efficiency* of a belt is affected by the direction in which it operates; a vertical belt being the least efficient and a horizontal belt the most efficient. It is also affected by the velocity and the tension.

80. Velocity:—Belting is said to give out its *maximum efficiency* at velocities between 5000 and 6000 feet per minute. The best current practice for high speed belts, however, is from 4000 to 4500 F. P. M. With a given velocity of belt and a known horse power, we can assume a working stress per square inch of section and solve for the square inches of belt area.

81. Fibre Stress:—Concerning the *working strength* of leather belting many references might be quoted, chiefly those in Kent, Pages 876-887. These references however, show such a lack of uniformity that it becomes largerly a matter of the judgment of the designer. Tests of belting show an ultimate strength of about 4000 pounds per square inch, which, with a factor of safety of 10, gives $f = 400$ pounds per square inch as a maximum working stress. This figure varies in practice from 250 to 400 pounds and agrees with a turning effort, p per square inch of section, between 180 and 290 pounds. Kent seems to favor a turning force of 275 pounds per square inch, which is equivalent in a single belt to 52 pounds and in a double belt to 86 pounds per inch of width. These figures seem to give results that agree well with current practice in cemented or endless belts.

82. Thickness:—The thickness of belting is usually

$$\text{single belt} = \frac{3}{16} \text{ inch.}$$

$$\text{double belt} = \frac{5}{16} \text{ inch.}$$

$$\text{triple belt} = \frac{7}{16} \text{ inch.}$$

83. Width:—The width of a belt is always determined from the horse power formula. The simplest way of obtaining this is as follows:

Application:—Given a pulley running at 300 R. P. M. and transmitting 100 H. P. with a belt speed of 4000 F. P. M. What is the diameter of the pulley and the width of the belt?

$$V = 4000 = \pi D N; \; D = 4.24' = 51''$$

Take the ultimate strength of belting at 4000 #□″ and the factor of safety at 10, then $f = 400$#□″. Let $T_2 = 3\ T_1$, then $p = 400 - 133 = 267$#□″. This is equivalent to 50 pounds per inch in width of single belt and 84 pounds per inch in width of double belt and agrees very closely with Kent's recommendations. From the formula

$$H. P. = \frac{P V}{33000}; \; 100 = \frac{P \times 4000}{33000}; \; P = 825 \text{ pounds.}$$

$$\frac{P}{p} = \frac{825}{267} = 3.1 \square'' = \begin{cases} 16.5'' \text{ single belt.} \\ 10'' \text{ double belt.} \end{cases}$$

84. Another very satisfactory formula if it is desired to take into account the centrifugal tension and the arc of contact of the belt is that given in Kent, Page 878 by Nagle. (For proof of this formula see Par. 93.

$$H. P. = C\ v\ t\ w \left(\frac{f - .012\ v^2}{550}\right) \qquad (1)$$

$C = 1 - 10^{-.00758\,\phi\,\alpha}$

α = degrees of belt contact.

ϕ = coefficient of friction taken at say .4

w = width of belt in inches.

t = thickness of belt in inches.

v = velocity in feet per second.

f = stress on belt per square inch of section.

$$\text{Take } f = \begin{cases} 275 \text{ for laced belts.} \\ 400 \text{ for lapped and riveted belts.} \end{cases}$$

Table VIII.—Values of $C = 1 - 10^{-.00758\,\phi\,\alpha}$ for different arcs of contact.

ϕ = Coeff. of Friction	DEGREES OF CONTACT = α										
	90	100	110	120	130	140	150	160	170	180	200
.15	.210	.230	.250	.270	.288	.307	.325	.342	.359	.376	.408
.20	.270	.295	.310	.342	.364	.386	.408	.428	.448	.467	.503
.25	.325	.354	.381	.407	.432	.457	.480	.503	.524	.544	.582
.30	.376	.408	.438	.467	.494	.520	.544	.567	.590	.610	.649
.35	.423	.457	.489	.520	.548	.575	.600	.624	.646	.667	.705
.40	.467	.502	.536	.567	.597	.624	.649	.673	.695	.715	.753
.45	.507	.544	.579	.610	.640	.667	.692	.715	.737	.757	.802
.55	.578	.617	.652	.684	.713	.739	.763	.785	.805	.822	.853
.60	.610	.649	.684	.715	.744	.769	.793	.813	.832	.848	.877
1.00	.792	.825	.853	.877	.897	.913	.927	.937	.947	.956	.969

Appplying this formula to the above problem we have for $\alpha = 180°$ and $\phi = .4$

$$100 = .715 \times 66.66 \times \tfrac{5}{16} w \left\{ \frac{400 - .012\ (66.66)^2}{550} \right\}$$

w = 10.5 inches for double belt.

or w = 17.7 say 18 inches for single belts.

85. A good short cut rule for finding the capacity of belting appeared in Power, November, 1903, Page 621.

RULE—"*To find the Horse Power which a given belt will transmit;* Multiply the diameter of either pulley in feet by its revolutions per minute and by the width of the belt in inches. Multiply the above product by a value ranging from 3 for light single laced belts to 8 for heavy double cemented belts and point off three places from the final product.' If C = constant to be chosen, then

$$\frac{C \times D \times R.P.M. \times w}{1000} = H.P.$$

Applying this formula to the above problem we have, if the belt is to be double thickness and is to have cemented joint

$$w = \frac{100 \times 1000}{4.24 \times 300 \times 8} = 10 \text{ inches.}$$

The chief difficulty in this will be to assume the proper constant. Experience will be necessary in determining this constant, but when it is fully established the rule becomes one of easy application.

All of the above belting rules show the best practice in high speed belting, but if applied to shop conditions they must be modified decidedly. It must be remembered that most shop belts are laced or hooked, resulting in a much weakened belt. The strength of a laced, or hooked joint may be taken from 30 to 50% of the value of a solid belt, consequently, if the same factor of safety be used, the value of f will be correspondingly smaller. In Nagles formula $f = 400$ for solid or cemented belts, and 275 for laced or hooked belts. No real definite ratio can be laid down governing shop belts, but the designer must take into account all the conditions, and make every belt a special case.

86. The sizes of belts leading to machines and counters must conform to the pulley sizes as specified by the manufacturers. For the average machine shop with a variety of lathes, planers, milling machines, shapers, drill presses and the like, the belts range from 2 to 4 inches single thickness. With the introduction of the new self hardening tool steel, the cutting speeds, and the weight of metal removed per hour are being increased to such an extent as to require a revision of former sizes. This, of course, is due to the increased power required to run the machines.

The speed of belts leading to counters and machines varies from 400 to 1000 F. P. M.; this is far below the most efficient velocity, but is made necessary because of the limitations on the pulley diamaters.

Belts are manufactured according to the following widths.

1" to 3" by 1/4 inch variations.
3" to 7" by 1/2 inch variations.
7" to 24" by 1 inch variations.
24" to 48" by 2 inch variations.

87. Perforated Belts:—The rapid approach of a belt to a pulley forms an air cushion which reduces the friction of the belt on the pulley. To avoid this, belts are sometimes perforated. Link belts have this advantage with the additional advantage of flexibility.

Perforated pulley rims are very common and serve the same purpose as perforated belts.

88. Belt Fastenings:—Cement, lacing leather, wire and hooks. For an endless belt, cement and rivets, cement and stitching, cement and shoe pegs, or cement alone are used.

It is always advisable to have a cemented joint on a high speed belt. Proper arrangement however, must be made for taking up the stretch.

Never throw a wide belt onto a pulley from the side, if it can be avoided, there is danger of unduly stretching that side. Cement the belt while in place over the pulleys.

The total stretch of new leather belting is from 6 to 8% of the original length.

89. Relation Between the Tension in a Belt and the Pressure on a Pulley:—In any belt at rest, Fig. 20, let T = tension in pounds per inch of width in each line, and p = normal pressure on the pulley rim per inch of width, then,

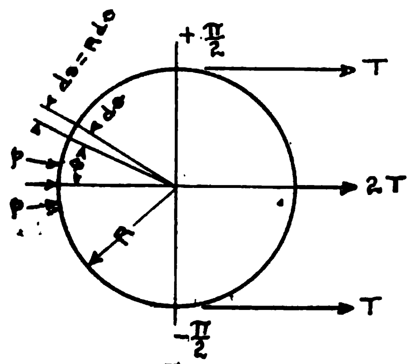


FIG. 20.

$$2\,T = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} p\,ds\cos\theta = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} p\,R\cos\theta\,d\theta$$

$$2\,T = p\,R\Big[\sin\theta\Big]_{-\frac{\pi}{2}}^{\frac{\pi}{2}} \quad \text{and} \quad p = \frac{T}{R} \qquad (2)$$

90. Relation Between the Tensions of the Tight and Loose Sides of a Working Belt in Terms of the Arc of Contact:—Let T_2 and T_1 be the tensions, respectively, of the tight and loose sides of the belt; also let Φ be the coefficient of friction, B the arc of contact in π measure, and a the arc of contact in degrees, then from Fig. 21

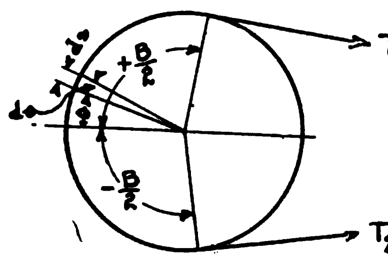


FIG. 21.

$$d\,T = d\,F = \Phi\,p\,d\,s = \Phi\,\frac{T}{R}\,R\,d\,\theta$$

$$\int_{T_1}^{T_2}\frac{d\,T}{T} = \int_{-\frac{B}{2}}^{\frac{B}{2}}\phi\,d\theta \text{ and } \Big[\log_e T\Big]_{T_1}^{T_2} = \phi\Big[\theta\Big]_{-\frac{B}{2}}^{\frac{B}{2}}$$

$\log_e \dfrac{T_2}{T_1} = \Phi\,B.$ but since $B = 3.1416\;a \div 180$

we have $\log_e \dfrac{T_2}{T_1} = .01745\;\Phi\,a$ and $\log \dfrac{T_2}{T_1} = .00758\;\Phi\,a$

$$\text{from which we get } \frac{T_2}{T_1} = 10^{.00758\,\phi\,a} \qquad (3)$$

91. Horse Power Formula in Terms of the Maximum Pull on the Belt, T_2 and the Arc of Contact:—Having given the formulas

$$\text{(A), H. P.} = \frac{P\,V}{33000};\ \text{(B)},\ P = T_2 - T_1;\ \text{and (C)},\ \frac{T_2}{T_1} = 10^{.00758\phi\,a}$$

$$\text{obtain, } T_1 = \frac{T_2}{10^{.00758\,\phi\,a}};\ P = T_2\,(1 - 10^{-.00758\,\phi\,a}) = T_2\,C$$

$$\text{and finally H. P.} = \frac{T_2\,C\,V}{33000} \qquad (4)$$

92. Horse Power Formula in Terms of the Thrust on the Bearing, 2 T. and the Arc of Contact:—In addition to the three formulas (A), (B) and (C), assume (D), $2\,T_0 = T_2 + T_1$ (See Church Par. 171), and obtain

$$T_2 = T_1\,(10^{.00758\;\phi\,a});\ 2\,T_0 = T_1\,(1 + 10^{.00758\;\phi\,a});$$

$$T_1 = \frac{2\,T_0}{1 + 10^{.00758\,\phi\,a}};\ P = 2\,(T_0 - T_1) = 2\,T_0\,\Big(1 - \frac{2}{1 + 10^{.00758\,\phi\,a}}\Big)$$

and finally H. P. $= \dfrac{2\, T_0\, V}{33000}\left[\dfrac{10^{.00758\,\phi\,\alpha} - 1}{10^{.00758\,\phi\,\alpha} + 1}\right]$ (5)

This formula may be especially useful in the design of machines for power measurements.

93. Horse Power Formula in Terms of T_2, Arc of Contact, and Centrifugal Tension:—The centrifugal force of any piece of belting swinging around the centre of the pulley, causes it to push away from the pulley rim with a unit force p_c. This increases the value T_2 as previously given. From a former proof:

$$p_c = \frac{T_c}{R} = \frac{M\,v^2}{R}$$

where M = the mass of the leather and v = velocity in feet per second. Then if W = weight of a piece of leather 1 x 1 x 12 inches (56 ÷ 144 = .388 pounds), and V = velocity in feet per minute,

$$T_c = \frac{W}{g}\,\frac{V^2}{3600} = \left[\frac{.012\,V^2}{3600}\right] \qquad (6)$$

Again, if w = width of belt in inches, and t = thickness in inches, the pull on the tight side of the belt will be increased by the centrifugal tension and

$$f\,w\,t = T_2 + \frac{.012\,V^2}{3600}\,w\,t. \text{ and}$$

$$T_2 = w\,t\left[f - \frac{.012\,V^2}{3600}\right]$$

Then by substituting in (4) we have.

$$\text{H. P.} = \frac{C\,V\,w\,t}{33000}\left[f - \frac{.012\,V^2}{3600}\right] \qquad (7)$$

By comparison it will be seen that (7) is the same as (1).

94. Velocity at Which a Belt will Transmit its Maximum Power:—In (7), substitute $f = 400$, then if the first differential coefficient is taken equal to zero

$$\frac{d\,H\,P}{d\,V} = \frac{C\,w\,t}{33000}\left(400 - \frac{.036}{3600}\,V^2\right) = 0.$$

$$V^2 = \frac{400 \times 36000}{.036} = 40{,}000{,}000.$$

$V = 6300$. Feet per minute.

95. Velocity at Which the Tension, Due to the Centrifugal Force, will Equal the Working Strength:—

In (7) let $400 = .012\,V^2 \div 3600$ then
$V = 11000$ Feet per minute.

96. Velocity at Which a Belt Will Break, Due to the Centrifugal Tension:—

In (7) let $4300 = .012\,V^2 \div 3600$ then $V = 35900$ Feet per minute.
This value is about 50 per cent greater than the bursting speed of solid cast iron pulley rims.

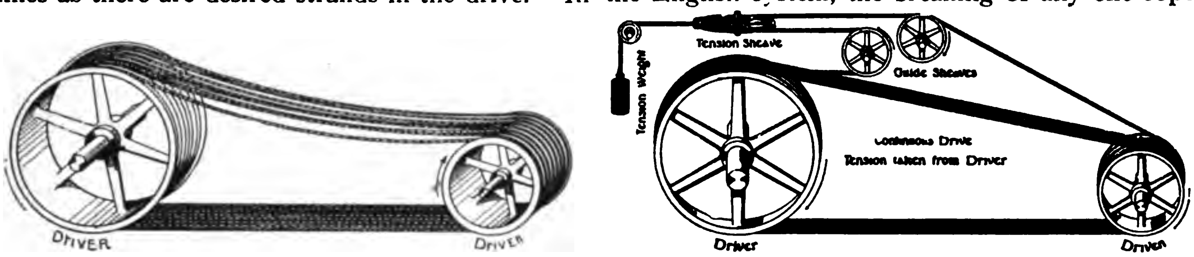
ROPE TRANSMISSION.

97. Materials:—Power ropes are made from wires, manilla, hemp, cotton and rawhide. Wire ropes are used in cable work for hoists, conveyors and tramways, and fibrous ropes are preferred for pure power transmission. Of the kinds mentioned, *manilla* rope is the most common.

98. Velocity, Fibre Stress and Size of Ropes:--The *velocity* of a transmission rope should be about 5000 feet per minute for maximum efficiency. The *fibre stress* should be about 255 pounds per square inch. This is usually spoken of in terms of the diameter of the rope, as 200 d^2. Rawhide may be taken 250 d^2. The *size* of the rope may vary between ⅞ inch for small powers and 2 inch for large powers. Probably the most common sizes are 1⅛ and 1½ inch.

The rope size may be governed, in some cases by the diameter of the smaller sheave, i. e. the *diameter* of the *smaller sheaves* should be *at least* 36 *d*.

99. Systems:—There are two systems of rope drives, Fig. 22. *A*, the *English* system has a number of ropes running in parallel over the same pulleys, each rope acting independently of the others; and the *American* systems which has but one rope looped continuously around the pulleys, as many times as there are desired strands in the drive. In the English system, the breaking of any one rope



A B

FIG. 22.

does not seriously disable the system, while in the American, it would shut down the plant. On account of this, a close inspection of the ropes must be made at regular intervals to guard against accidents. The method of tightening the rope in the latter system, however, gives it an advantage over the former, in that there is less slippage between the ropes and the groove and consequently less wear on the ropes.

For details of design see "Flather's Rope Drives," and "A Little Blue Book on Rope Transmission" by the American Manufacturing Company.

100. Simple Horse Power Formula:—In rope drives having slow speeds, and in such designs as do not require extreme accuracy, the following formula will be found satisfactory. Let $T_2 \div T_1 = 2$, $V =$ velocity of rope in feet per min., and $N =$ number of ropes in the drive, then $P = T_2 \div 2 = 200\, d^2 \div 2$ and

$$\text{H. P.} = \frac{100\, d^2\, N\, V}{33000} \qquad (8)$$

101. Relation Between the Tensions of the Tight and Loose Sides of a Working Rope, in Terms of the Arc of Contact:—Let θ be the angle of the groove, Fig. 23, (usually taken at 45°), then $2\, \Phi p_1 = (\Phi \operatorname{cosec} \theta \div 2) p$. Let $(\Phi \operatorname{cosec} \theta \div 2) = \mu =$ equivalent coefficient of friction and substituting in (3); we have

$$\frac{T_2}{T_1} = 10^{\,.00758\, \mu\, a} \qquad (9)$$

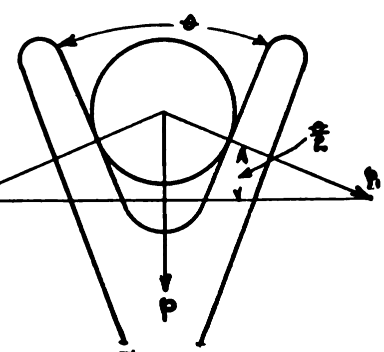


FIG. 23.

102. Horse Power Formula in Terms of the Maximum Pull on the Ropes, T_2, and the Arc of Contact:—From (9), as in Par. 89, we have $T_2\,(1 - 10^{-.00758\, \mu\, a}) = T_2\, K$ then

$$\text{H. P.} = \frac{T_2\, K\, V}{33000} \qquad (10)$$

103. Horse Power Formula in Terms of T_2, Arc of Contact, and Centrifugal Tension:—As in belting we have, if the weight of one cubic inch = .034 pounds.

$$T = \frac{W}{g}\,\frac{V^2}{3600} = \frac{.32\, d^2\, V^2}{32.2 \times 3600} = \frac{.01 d^2\, V^2}{3600} \qquad (11)$$

If T_2 = tension in the ropes on the tight side, not counting centrifugal tension, then the maximum fibre stress in the ropes under the new condition will be

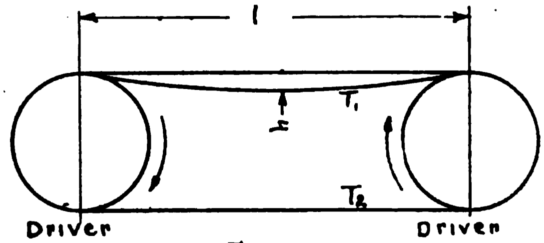


FIG. 24.

$$f = T_2 + \frac{.01\, d^2\, V^2}{3600} \text{ or } T_2 = 200\, d^2 - \frac{.01\, d^2\, V^2}{3600} \quad (12)$$

$$\text{then H. P.} = \frac{K\, d^2\, V}{33000}\left(200 - \frac{.01\, V^2}{3600}\right) \quad (13)$$

104. Relation Between the Tension and the Sag in a Rope or Belt:—The amount of sag in a flexible cord may be used in determining its tension. A rope is very nearly a flexible cord and will follow the same laws closely. The curve taken by a perfectly flexible cord hanging over pegs is a *catenary*, and is represented by

$$y = \frac{c}{2}\left[\left(e^{\frac{x}{c}} + e^{-\frac{x}{c}}\right) - c\right]$$

Unwin, Page 422. Where e is the base of the naperian system of logarithms. Referring to Fig. 25 this becomes

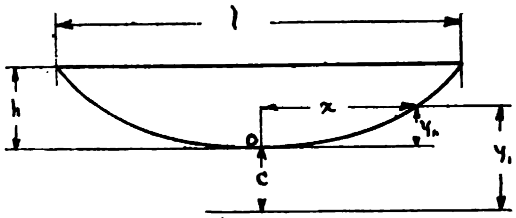


FIG. 25.

$$y + c = y^1 = \frac{c}{2}\left(e^{\frac{x}{c}} + e^{-\frac{x}{c}}\right) \quad (14)$$

If (14) be written in a decreasing series and collected to four terms

$$y' = \frac{c}{2}\left[2 + \frac{2\,x^2}{2\,c^2}\right] = c + \frac{x^2}{2\,c} \quad (15)$$

To determine c when $x = l \div 2$, $y' = c + l^2 \div 8\,c = c + h$ and $c = l^2 \div 8\,h$ (16)

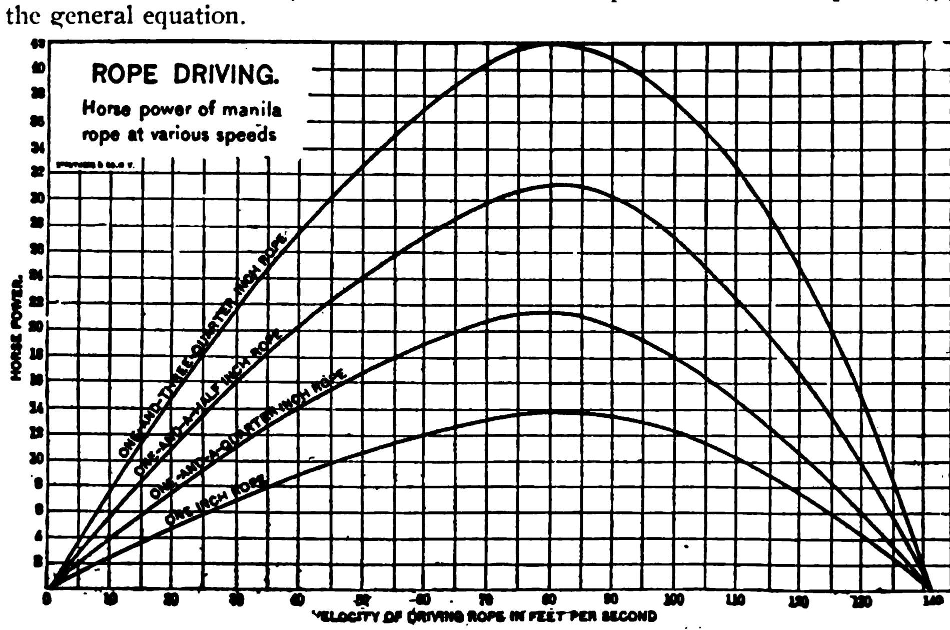
It is the property of this curve that the tension at any point of the curve equals the weight of a portion of the rope having a length y', measured to the point selected. Expressed as an equation, $T = W\,y'$. From this we obtain

$$T = W\,(c + h) = W\left[\frac{l^2}{8\,h} + h\right] \quad (16)$$

Where l = length of span in feet and h = deflection of the curve in feet. This equation is sometimes written

$$T = W\,\frac{l^2}{8\,h}$$

In the above, calculate T as T_1 of the slack side of the rope and then find T_2 from (9). This will give P in the general equation.



FLY WHEELS AND PULLEYS.

105. Fly Wheel Pulleys:—In designing a pulley, the following things must be known: (1), the velocity of the rim; this is given in F. P. M. of belt velocity or in diameter and R. P. M. of pulley; (2), the Horse Power transmitted; (3), the Weight of the pulley rim; (4), Single or double belt. Having given the above data with the knowledge of the use to which the pulley will be put, the following would be the method of procedure.

Given an engine pulley with D = 50″; R. P. M. = 250; H. P. = 75; weight of rim = 650#. Find the width of the belt and of the pulley rim and all sizes relating to the rim, arms, hub, shaft, and fastening.

106. Rim:—Allowing an arc of contact of 180° and a coefficient of friction of .4 we have, Par. 84, w = 9.4, say 10 inches. Taking the pulley rim one-half inch wider than the belt gives the width of the pulley face = 10.5 inches.

Let the thickness of the rim be *t* inches, then the weight of the rim becomes

$$W = \pi\ (D - t)\ .26\ t\ w.$$
$$650 = 3.1416\ (50 - t)\ t \times 10.5 \times .26$$
$$t = 1.56''$$

NOTE.—This assumes that the radius of gyration extends to a point only half way between the inner and outer surfaces of the rim. This assumption is not theoretically correct, but for all practical purposes the error is so slight as to be negligible. See Low and Bevis, Par. 47. The above calculation for the thickness of the rim assumes that about all the weight of the wheel is centered in the rim and that the arms need not be considered. This is largely true in engine fly-wheel pulleys. See Church, Par. 106.

107. Arms:—Select the shape of the arm section and calculate the arm as a beam under flexure. If this section is oval as shown in Fig. 26, the formula becomes

$$\frac{P\,R}{N} = .05\ b^3\ f.$$

where P = tractive force of the belt.
N = number of arms.
b = breadth of the arm at the center of wheel if projected to that point.
f = allowable fibre stress of the metal, say 1000 #□″; taken low because of unknown stresses due to centrifugal force.

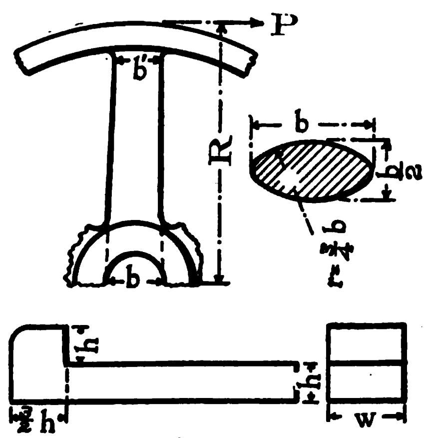


FIG. 26.

The sizes of the arm at the rim may be taken at about two-thirds of the hub sizes. In the above application for a six arm pulley we have,

$$b = \sqrt[3]{\frac{P.\,R.}{6 \times .05 \times 1000}} = 4.\text{ inches.}$$

From this value of *b* the other values may be obtained.

In designing pulley arms the size *b* at the center is sometimes taken equal to the diameter of the shaft and the other proportions as given above. This does not sufficiently account for all the straining actions of the belt and cannot be recommended further than as a check on the calculations.

Straight arms are preferred to curved arms and they should have good, well rounded fillets next to the hub and rim.

108. Shaft:—Since this pulley is to be used on an engine shaft, the diameter of the hole will be

$$d = 7.3\ \sqrt[3]{\frac{H.\,P.}{R.P.M.}} = 4.88,\text{ say } 4\tfrac{7}{8}\text{ inches. See Par. 63.}$$

109. Hub:—It is common practice to take the *diameter* of the hub as twice the diameter of the shaft, this would be 9¾ inches. The *length* of the hub should bear some relation to the width of the rim or to the shaft diameter. A ratio sometimes used is twice the shaft diameter. A more satisfactory figure would be from two thirds the width of the rim to the full width of the rim.

STRESSES IN FLY-WHEEL RIMS PRODUCED BY CENTRIFUGAL FORCE.

110. In any pulley moving at a high rotative speed there is a uniform expansion of the rim which tends to increase its diameter, thus producing a uniform tensional force in the rim section. There is also an increased length of the arm, but the expansion of the arms not being sufficient to equal the increased rim diameter, there is a bending moment produced in the rim between the arms. The uniform tensional stress and the bending stress may each be determined for any section of the rim and then combined as follows into a fibre stress f.

111. Centrifugal Tension (T):—The magnitude of the centrifugal force is given by the equation

$$C. F. = \frac{W v^2}{g R} \tag{1}$$

where W = weight in pounds, v = velocity in F. P. S., R = radius in feet, g = 32.2. Let t = thickness of rim in inches, w = width of rim in inches, l = length of unit arc in inches; then if one cubic inch of cast iron weighs .26 pounds, we have

$$W = w \times t \times l \times 0.26$$

For a unit section parallel with the rim, 1 inch × 1 inch this becomes

$$W = 1 \times t \times 1 \times 0.26 = .26\, t$$

substituting this value of W in (1) gives C F in pounds per square inch as

$$C. F. = p = \frac{0.26\, t\, v^2}{32.2\, R} \tag{2}$$

For a cylinder subjected to an internal pressure of p pounds per square inch we have $f = p\, D \div 2t$, where f = tension produced in the rim in pounds per square inch, D = diameter in inches and t = thickness in inches. Substituting for p the value given by (2) we have

$$f = \frac{0.26\, t\, v^2 \times 2 \times 12 \times R}{32.2\, R \times 2 \times t} = \frac{v^2}{10.3} \tag{3}$$

from which we see that if we neglect the effect of the arms, the stress produced by the centrifugal force in a pulley rim varies as the square of the velocity and does not depend on the dimensions of the pulley. This stress in the rim will be called the centrifugal tension and is the stress that is commonly taken into account by designers.

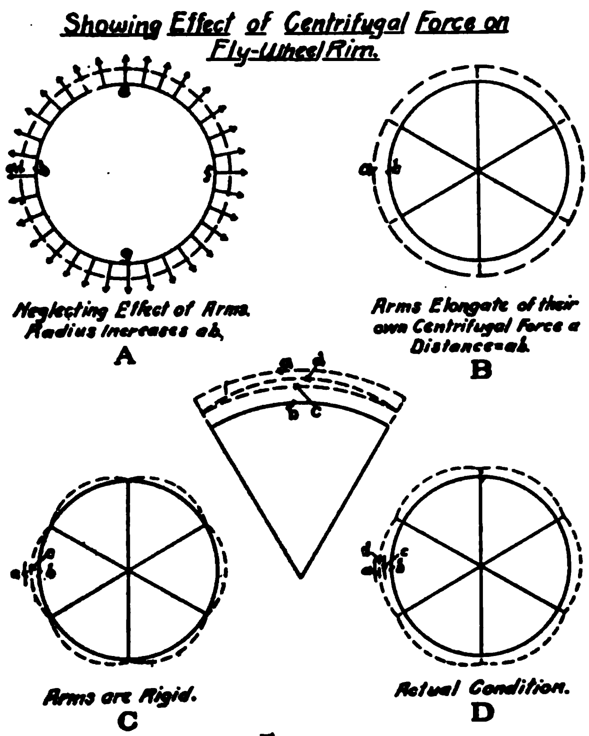


FIG. 27.

112. The stresses *actually existing* in the rim when in motion are very difficult to determine. It is safe to assume that they are as follows:

(1) A direct tensile stress which is a portion only of the centrifugal tension.

(2) Stresses due to the bending of the portion of the rim between two adjacent arms.

To explain more fully, suppose that *befg* Fig. 27 A, is the center of the rim of a pulley when at rest. If the pulley is put in motion and we neglect the effect of the arms, centrifugal tension will increase the circumference as shown by the dotted line; but if we take into account the arms and regard them as allowing the full expansion of the rim, it is evident that they must lengthen an amount *ab* as shown at B. On the other hand if the arms were inextensible we would have the condition of a beam uniformly loaded as shown at C. The actual shape of the rim, however, will be somewhere between B and C as shown in D, where the arms elongate a distance *bc* and the rim bends an amount *cd*. Hence the statement for the stresses which actually exist in the rim of a pulley as given above when put in a formula is:

$$f = \frac{T}{A} \pm \frac{M}{Z} \quad \begin{cases} - = \text{outside of rim.} \\ + = \text{inside of rim} \end{cases} \tag{4}$$

where f = stress in pounds per square inch at any point.
T = tension in rim due to centrifugal force, in pounds.
A = area of rim section in square inches.
M = bending moment.
Z = Modulus of section.

To determine T and M it is necessary to first find the pull F exerted on each arm such that F ÷ 2 is the shear at the point of support. It is very evident that

$$F = C \frac{W}{g} v^2 \tag{5}$$

where C is a constant to be determined. Now it can be shown analytically* that when W = weight of a cubic foot of iron = 450 pounds and g = 32.2 the value of C is

$$C = \frac{1}{3} \left\{ \frac{3 - \left(\frac{r_1 - r_2}{R}\right)^2 \left(\frac{r_1 + \frac{1}{2} r_2}{R}\right)}{\frac{1}{A'_1}\left(\frac{r_1 - r_2}{R}\right) + \frac{1}{2 A' a}} \right\} \tag{6}$$

r_1 = distance in feet from center of hub to outer end of arm.
r_2 = radius of hub in feet.
R = distance from center of rim to center of hub in feet.
A′ = area cross section of rim in square feet.
A_1' = area cross section of arms in square feet.

a = ½ angle between arms in π measure.
Φ = angle between arm and a variable point.

Using the value of F thus determined it gives

$$T = \frac{W}{g} A' v^2 - \frac{F}{2} \frac{\cos(a - \Phi)}{\sin a} \tag{7}$$

$$M = \frac{FR}{2}\left(\frac{1}{a} - \frac{\cos(a - \Phi)}{\sin a}\right) \tag{8}$$

and f the maximum stress in the rim becomes known by substituting in equation (4). Applying the above to a wheel as shown in Fig. 28 we have from formula (6), if v = 60 F. P. S.

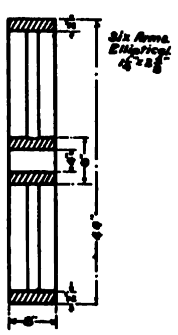


FIG. 28.

C = .0189

Substituting this value in (5) we have

F = 951.

Substituting this value in (7) for each of the angles 0, 10, 20 and 30 degrees we have

	0°	10°	20°	30°
T =	4766	4696	4653	4639
and $\frac{T}{A}$ =	298	293	291	289

Substitute the value F as given above, also in (8) and obtain

	0°	10°	20°	30°
M =	146.29	24.63	—50.43	—75.7
$\frac{M}{Z}$ =	331	55.	—113.	—170

*For complete analysis see Vol. XVI Proceedings A. S. M. E. "Stresses in Rims and Rim-joints of Pulleys and Fly Wheels." Lanza.

If now, we substitute the values $\frac{T}{A}$ and $\frac{M}{Z}$ in equation (4) it gives

	0°	10°	20°	30°	
$f =$	629	348	178	119	Inside fibres.
$f =$	—33	237	404	461	Outside fibre.

Plotting the values $\frac{T}{A}$, $\frac{M}{Z}$ and f for the inner and outer fibres of the rim we see Fig. 29 that at

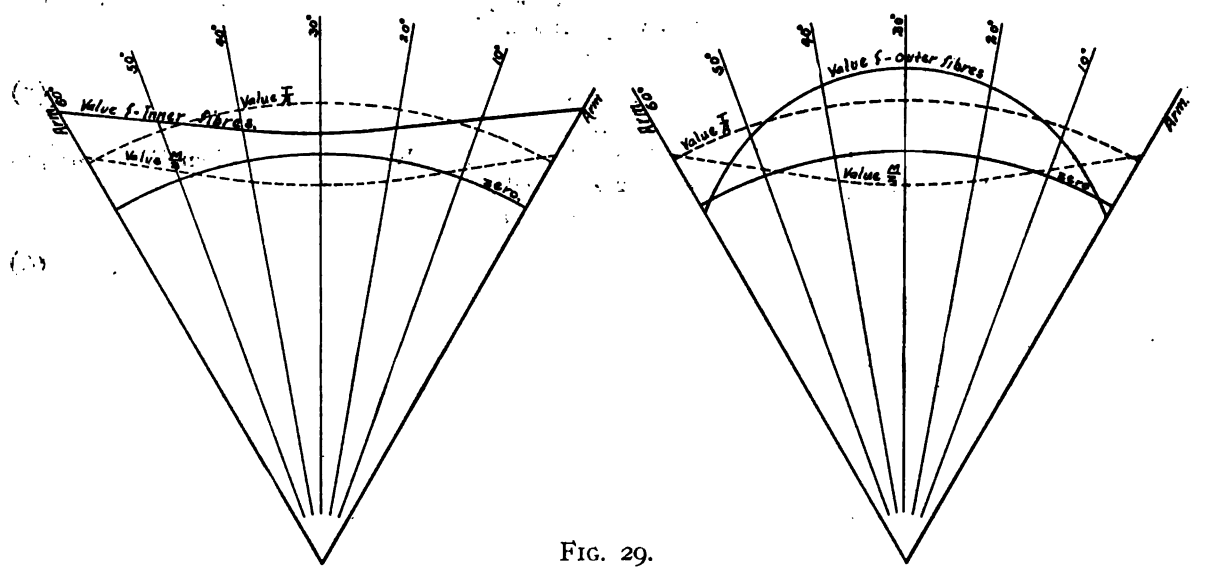


FIG. 29.

about thirteen degrees from each arm the stresses in the outer and inner fibres are equal. It is evident then that the rim joints of sectioned rims might well be located at these points.

113. Forms of Pulleys; Rims:—Pulley rims and belt-fly-wheel rims are usually made of one of the forms A, B and C, Fig. 30. Under favorable conditions with a thick rim section A would be satisfactory; but if the rim is wide and not very thick the cross bending strain on the rim due to its centri-

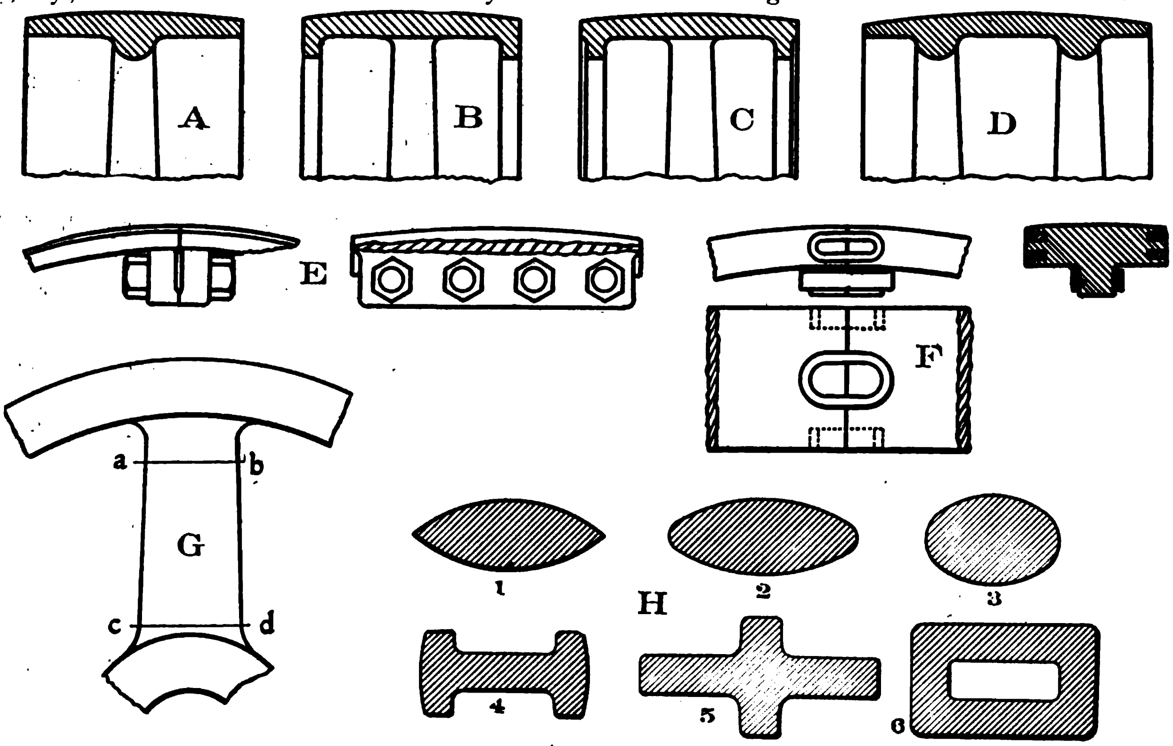


FIG. 30.

fugal force may endanger it and flanges as at B and C are added to support the outer edge. Where the rim is of unusual width, two sets of arms may be provided as at D.

Solid rims have unknown stresses in the metal due to the cooling action in the mold. These may be somewhat relieved by making the rim in sections and bolting the sections together. Other advantages are to be gained by sectioning the pulleys; the chief ones being, that of handling in its production and shipping, and the ease of adjustment to the shaft without removing the shaft from its bearings.

Two common methods of fastening the sections together are shown in E and F, Fig. 30. E is the simplest and is more often found than F but it is not so strong. In the Trans. A. S. M. E. Vol. 20, will be found a report of a series of experiments by Prof. Benjamin on the explosion of pulleys running at high speeds in which he brings out the following conclusions:

(1) Fly wheels with solid rims, of the proportions usual among engine builders and having the usual number of arms, have a sufficient factor of safety at a rim speed of 100 feet per second if the iron is of good quality and there are no serious cooling strains. In such wheels the bending due to the centrifugal force is slight and may be disregarded.

(2) Rim joints midway between the arms are a serious defect and reduce the factor of safety very materially. Such joints are as serious mistakes in design as would be a joint in the middle of a girder under a heavy load.

(3) Joints made in the ordinary manner, with internal flanges and bolts, are probably the worst that could be devised for this purpose. Under the most favorable circumstances they have only about one fourth the strength of a solid rim and are particularly weak against bending. In several joints of this character on large fly wheels, calculation has shown a strength less than one fifth that of the rim.

(4) The type of joint known as the link or prisoner joint is probably the best that could be devised for narrow rimmed wheels not intended to carry belts, and possess when properly designed, a strength about two thirds that of the solid rim.

114. Rope pulley rims take the forms shown by Fig. 30. A and B are typical sections of transmission pulley rims for fibrous ropes; C, is a section of a rim for wire rope transmission; and D, is the section of an idler pulley rim. Attention is called to the fact that the fibrous ropes *wedge* in the grounds, and wire ropes rest upon the bottom of the groove. Also, that the wire rope groove has a hard wood bottom.

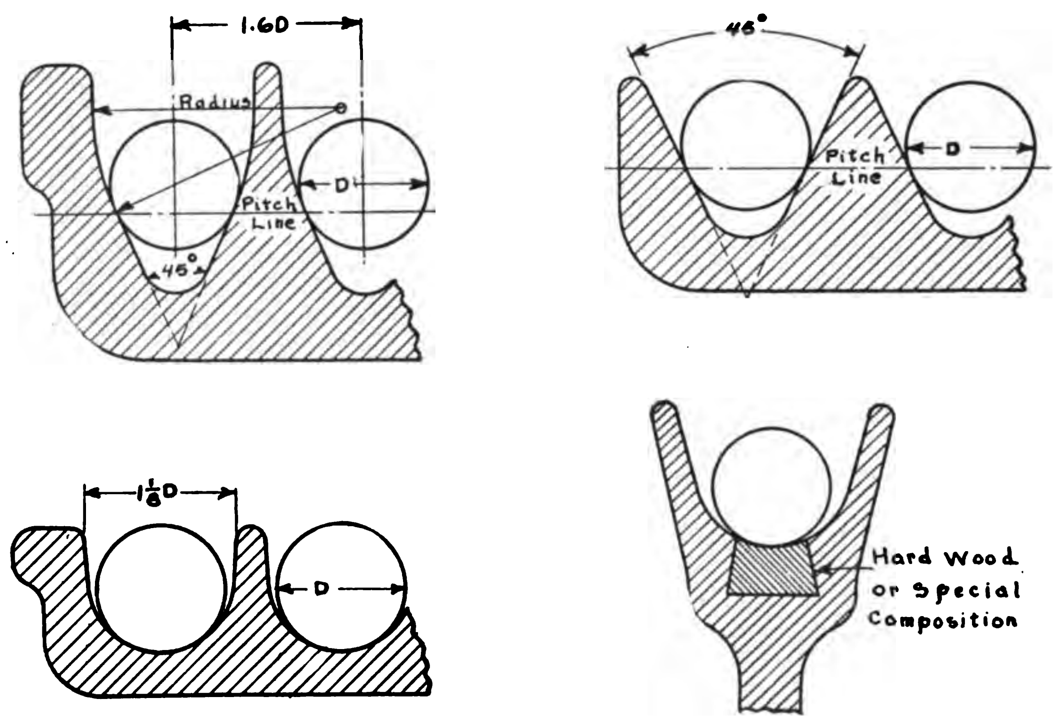


Fig. 31.

115. Arms:—The section of the arm near the rim and the section near the hub are always similar although the former is only about two-thirds the size of the latter. Of the sections shown in H, Fig. 30, 1 to 6, 2 is the one found in general use on medium sized wheels; large wheels having arms shaped like 4, 5 and 6. Sections 1 and 2 have an advantage over the others in a wind resistance. This in high speed wheels is a factor worthy of being considered.

116. Hubs:—Hubs in like manner with the rims are made either solid or sectioned. Many wheels have solid rims and split hubs, thus with the flexibility at the hub reducing the cooling strains in the wheel and providing an easy adjustment to the shaft. In Fig. 32, I shows a typical solid hub section, J shows a single cut hub and K a double cut hub.

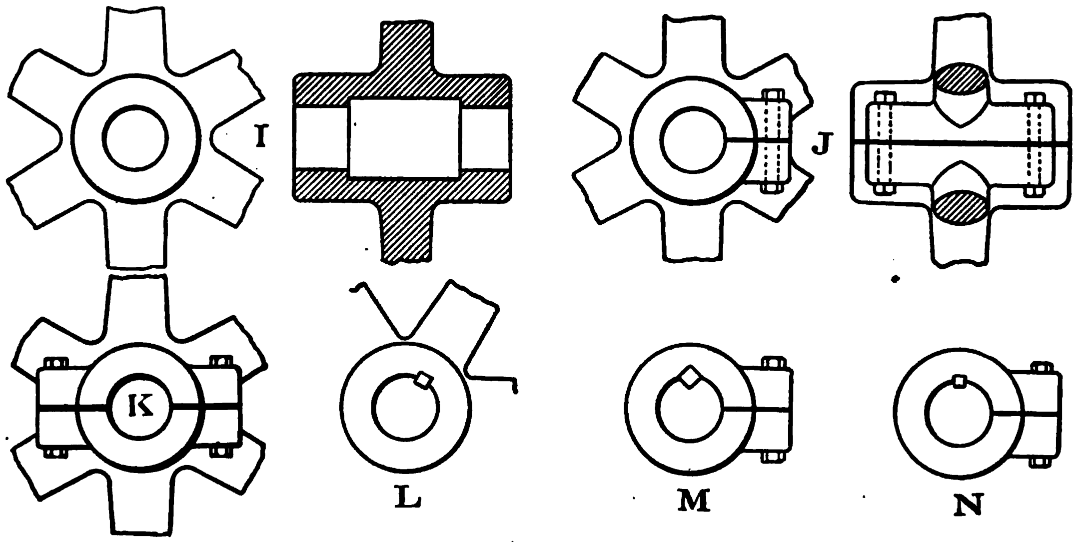


FIG. 32.

A sectioned hub makes it easy to remove the key and wheel from the shaft. In fitting keys to pulleys they are generally placed under an arm. In a solid hub any arm will do, but in split hubs a location should be taken approximately at right angles to the cut where the tension on the bolt produces a clamping action on the key between the hub and the shaft. Keys are sometimes set diagonally; this however, is not to be recommended.

117. Shafting Pulleys:—Pulleys for use on line and counter shafts are made of wood, cast iron and steel. Those used on machines are generally made of cast iron.

The wood pulley is generally split in two parts and bolted around the shaft, being held to the shaft by friction. Wood pulleys have found a large use in factories because of the ease with which they can be applied to or removed from the shaft. They can be had of any standard diameter and width of face. Any one pulley can be fitted to shafts of varying diameters by the use of bushings.

The cast iron pulley is the standard pulley. It is usually made solid although a split pulley can be obtained by special order. The following notes will apply to small, solid cast iron pulleys.

The diameter D usually varies by inches.

The width of face B = width of belt + ($\frac{1}{4}''$ to $\frac{1}{2}''$)

The thickness of the rim, t, $= \frac{D}{200} + \frac{1}{4}''$

Pulleys should be coned $\frac{1}{32}''$ to $\frac{1}{16}''$ in diameter for each inch of width, the narrower faces having the heavier cone.

Tight and loose pulleys used for shifting belts do not need coning.

Taper the inside of the rim and have good fillets between rim and arms.

The section of the arm can be calculated as shown in Par. 107. As a matter of fact, however, the arms of small pulleys are made much heavier than would be calculated from the forces involved. The chief reason for this is that pulleys may be used for any kind of service and must be designed to withstand the heaviest loads.

Concerning the number of arms in a pulley it may be said that in general, pulleys below 12 inches in diameter have 4 arms, and above 12 inches, 6 arms.

HUBS, KEYS AND COUPLINGS.

118. Standard Hubs:—The following hub sizes of various machine parts, representing the current practice of four manufacturing companies, was reported in the American Machinist, January 14, 1904.

Diameter of Hubs where d = Diameter of Shaft.

	Cast Iron.	Cast Steel.
Heavy, very great shock	2 d	1¾ d + ⅛″
Standard medium shock	1¾ d + ⅛″	1⅝ d + $\frac{3}{16}$″
Light, no shock	1⅝ d + ⅛″	1½ d + ¼″

Length of Hubs.

Truck wheels	2 d to 2¼ d	Gear wheels	1¾ d to 2¼ d
Hand wheels	1½ d to 2 d	Bearings	3 d to 4 d
Levers	1½ d	Pulleys	Face.

119. Keys:—A key for a large pulley should be made according to the shape shown in Fig. 26.

If d = diameter of shaft

then W = ⅕ d for a six inch shaft.

W = ¼ d for a two inch shaft.

h = ⅔ W to ¾ W for a six inch shaft.

h = ¾ W to W for a two inch shaft.

The key should *taper* ⅛ inch to $\frac{3}{16}$ inch per foot of length.

The following tables IX, concerning the size of hub keys and key seats, are in current use by reputable manufacturing firms.

TABLES IX.

DIAMETER OF SHAFT.	SIZE OF KEY SEAT.	
	WIDTH "A."	DEPTH "B."
Inches.	Inches.	Inches.
¾ to 1¼ inclusive	¼	⅛
1$\frac{5}{16}$ to 1¾ "	⅜	$\frac{3}{16}$
1$\frac{13}{16}$ to 2¼ "	½	¼
2$\frac{5}{16}$ to 2¾ "	⅝	$\frac{5}{16}$
2$\frac{13}{16}$ to 3¼ "	¾	⅜
3$\frac{5}{16}$ to 3¾ "	⅞	$\frac{7}{16}$
3$\frac{13}{16}$ to 4¼ "	1	½
4$\frac{5}{16}$ to 4¾ "	1⅛	$\frac{9}{16}$
4$\frac{13}{16}$ to 5¼ "	1¼	⅝
5$\frac{5}{16}$ to 5¾ "	1⅜	$\frac{11}{16}$
5$\frac{13}{16}$ to 6¼ "	1½	¾

STEEL KEYS IN SHAFTS.

Size of Shaft	$\frac{11}{16}$–$\frac{3}{4}$	$\frac{13}{16}$–$\frac{7}{8}$	$\frac{15}{16}$–1	1$\frac{1}{16}$–1$\frac{1}{8}$	1$\frac{3}{16}$–1$\frac{1}{2}$	1$\frac{9}{16}$–1$\frac{7}{8}$	1$\frac{15}{16}$–2$\frac{3}{8}$	2$\frac{7}{16}$–2$\frac{7}{8}$	2$\frac{15}{16}$–3$\frac{1}{2}$	3$\frac{9}{16}$–4	4$\frac{1}{16}$–4$\frac{5}{8}$	4$\frac{11}{16}$–5$\frac{1}{4}$
Size of Key	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$

120. Couplings:—Pieces of shafting are held together by couplings. Fig. 33 shows the common forms. The solid sleeve coupling is very little used except on very light work, the compression coupling and the double cone vise coupling are used on shafting lines, the flange coupling is used on heavy trans-

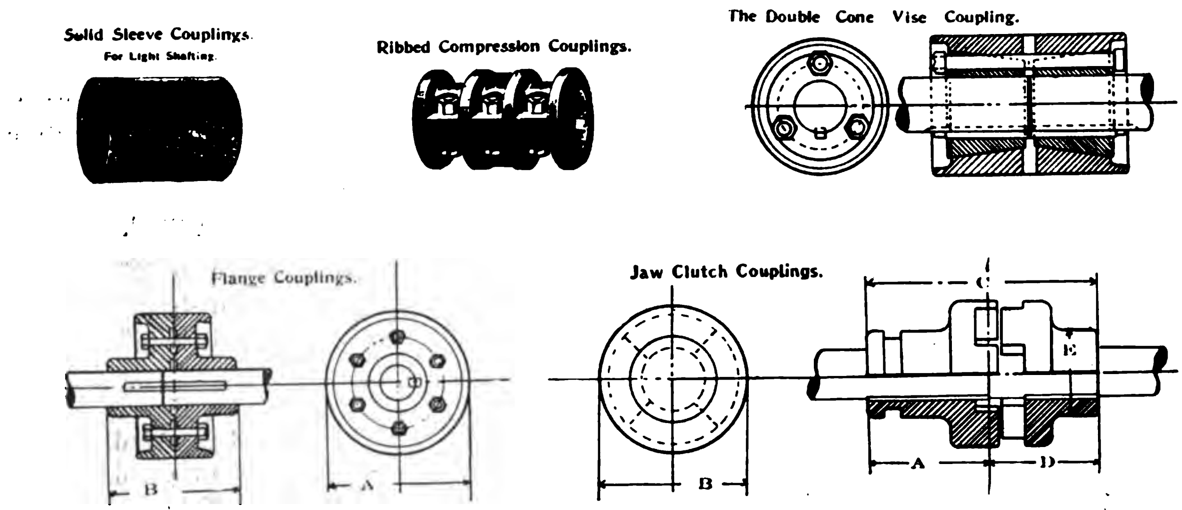


FIG. 33.

mission lines such as principal shafts and jack shafts, and the jaw clutch coupling is used on shafts requiring frequent disconnections. Tables X, from catalog data, will be of value in proportioning the various parts.

FLANGE COUPLING

Size of Shaft in Inches	A	B
1 3/16	6	4 1/8
1 7/16	6 5/8	5
1 11/16	7 1/4	5 9/16
1 15/16	7 13/16	6 1/8
2 3/16	8 3/8	6 5/8
2 7/16	9	7 1/4
2 11/16	9 1/2	7 3/4
2 15/16	10 1/8	8 5/16
3 3/16	10 11/16	8 7/8
3 7/16	11 1/4	9 3/8
3 11/16	11 7/8	9 7/8
3 15/16	12 3/8	10 1/2
4 3/16	13	11
4 7/16	13 1/2	11 5/8
4 15/16	14 1/4	12 5/8
5 7/16	15 7/8	13 3/4
5 15/16	17	14 7/8

THE DOUBLE CONE VISE COUPLING

Size of Shaft	Approx Diameter of Coupling	Approx Length of Coupling	Width of Key-Seat	Depth of Key-Seat	Length of Keys	Approx Weight Each in Pounds
Inches	Inches	Inches	Inches	Inches	Inches	
1 7/16	3 1/4	5 1/2	5/16	3/16	2 1/8	13
1 11/16	4 5/8	6 1/8	5/16	3/16	2 1/2	20
1 15/16	5 1/8	7 5/8	7/16	1/4	2 7/8	33
2 3/16	5 7/8	8 1/4	7/16	1/4	3 1/4	42
2 7/16	6 9/16	9 1/8	9/16	5/16	3 5/8	60
2 11/16	7	10	11/16	3/8	4	72
2 15/16	7 3/4	11	11/16	3/8	4 3/8	100
3 3/16	8 1/4	12	11/16	3/8	4 3/4	120
3 7/16	9 1/16	13	11/16	3/8	5 1/8	150
3 15/16	10	14 1/4	13/16	7/16	5 7/8	215
4 7/16	11 3/8	16	13/16	7/16	6 5/8	310
4 15/16	12 9/16	18	13/16	7/16	7 3/8	400
5 7/16	13 1/2	19 1/2	15/16	1/2	8 1/8	570
5 15/16	14 1/2	21	15/16	1/2	8 7/8	700

JAW CLUTCH COUPLING

Size	A	B	C	D	E
Inches	Inches	Inches	Inches	Inches	Inches
15/16	2	3	5 1/2	2	3 3/8
1 7/16	2 1/2	4 1/2	8	3 1/8	4 3/4
1 15/16	3 1/4	6	10 3/4	4	6
2 7/16	4 1/8	7 1/2	12 3/4	5	7 1/2
2 15/16	5 1/8	9	14 3/4	6	8 1/2
3 7/16	6 1/4	10 1/2	17 1/2	7	10
3 15/16	7	12	19 1/2	8	11
4 7/16	8	13 1/2	22	9	12 1/8
4 15/16	8 3/4	15	24	10	13 1/2
5 3/8	10 1/8	18	28 1/2	12	16
5 7/8	12	21	28 1/2	12	16 1/4

121. Counter Shafts:—Counter shaft journals and boxes differ materially from journals and boxes on line shafts. Counter shafts are sometimes made from pieces of cold rolled shafting without being

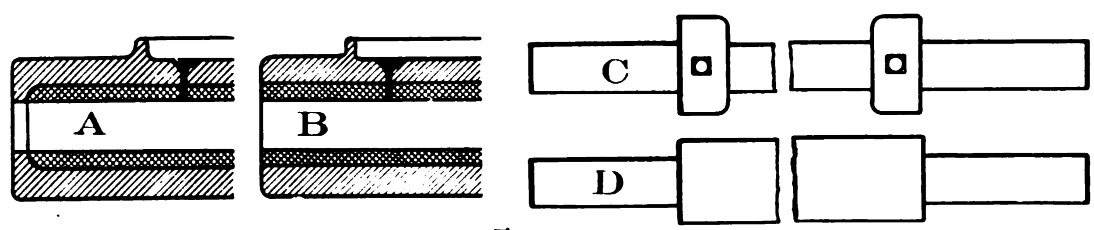


FIG. 34.

turned at the journals, the latter being located by set collars as at C, Fig. 34. The ordinary countershaft however, is turned smaller at the journals thus forming a shoulder and avoiding the necessity of a set collar, as at D.

Journal boxes for counter shafts are usually solid and babbitted, the babbitt being from 1/8 inch to 1/4 inch thick. If the box is to be used with a set collar the babbitt may come flush with the end of the casting as at B, but if the journal is shouldered the metal of the box should extend down *nearly to* the shaft as at A. In any case the end thrust of the shaft must be taken up *iron* to *iron*. Every counter shaft should have a small end play to avoid cutting the end of the box.

The following may be of value in finding the size of a counter shaft where it is to bear a certain relation to the width of the belt.

2″	Belt	$1\frac{3}{16}$″	shaft.
$2\frac{1}{2}$″	"	$1\frac{5}{16}$″	"
3″	"	$1\frac{7}{16}$″	"
$3\frac{1}{2}$″	"	$1\frac{9}{16}$″	"
4″	"	$1\frac{11}{16}$″	"
5″	"	$1\frac{13}{16}$″	"

122. Set Collars:—In all revolving shafts it is necessary to provide a means for taking up the end thrust of the shaft. This is done by set collars next the journal boxes. A, Fig. 35, shows their application to two boxes and B, to one. Collars should be located by the side of such boxes as are well braced, otherwise the hanger will vibrate and the belts will be set to swinging..

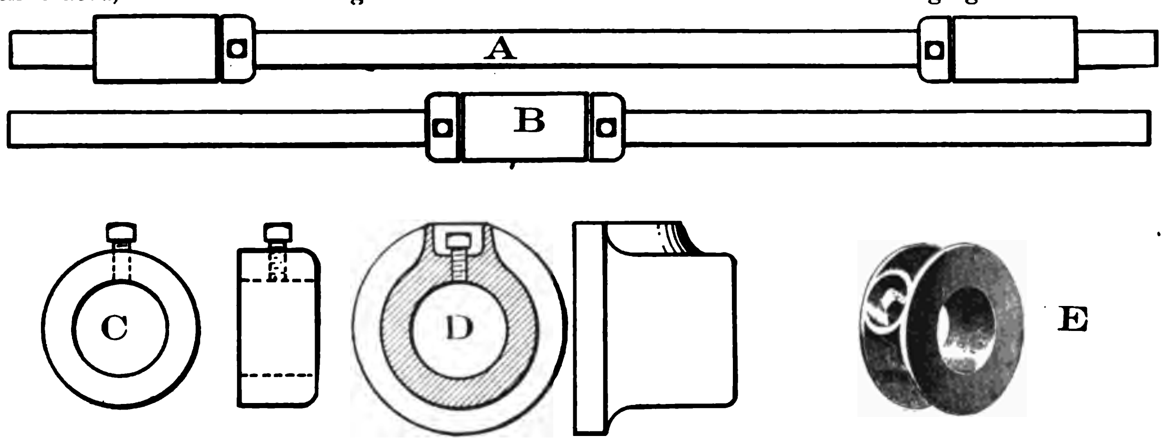


FIG. 35.

The simplest set collar is shown in C. This should not be used in a design because of the danger of the workman being caught with the screw. *Safety collars* as D and E should always be used.

123. Split Collars are made of the form similar to E. These can be removed or applied to the shaft without removing the shafting from the box.

TOOTHED GEARING.

124. Spur Gears:—Gear teeth are formed in three ways *i. e.,* pattern and machine molded, and machine cut. For large, slow moving or rough work the pattern molded tooth is used; for a somewhat better grade of work where the velocity is not too high and where an accurate mesh is not required the machine molded tooth is used; while for high grade machines requiring high velocities or a good fit between the teeth, cut gears are required. Table XI gives the ordinary proportions in use for the three kinds of teeth.

TABLE XI.

Kind of Tooth.	Addendum or Face.	Dedendum or Flank.	Clearance.	Total Height.	Width.	Space.
Pattern Molded.	.32 p	.38 p	.06 p	.7 p	.47 p	.53 p
Machine Molded.	.32 p	.38 p	.06 p	.7 p	.48 p	.52 p
Machine Cut.	.3183 p	.3581 p	.04 p	.6764 p	.5 p	.5 p

The two forms of gear teeth in general use are the *epicycloid* and the *involute.* The latter is preferred by many manufacturers. The following sizes for the involute tooth are used by the respective companies.

Brown Hoisting and Conveying Co.

Addendum $= A$; dedendum $= B$; radius at root of thread $= R$; width of tooth $= T$; $D.\ p =$ diametral pitch, and $p =$ circular pitch.

$$\textit{Cast tooth,}\quad A = \frac{.75}{D.\ p.} \qquad B = \frac{1}{D.\ p.} \qquad R = B - A \qquad T = .46\,p.$$

$$\textit{Cut tooth.}\quad A = \frac{.75}{D.\ p.} \qquad A + B = \frac{1.657}{D.\ p.} \qquad R = \frac{.157}{D.\ p.}$$

Wellman, Seaver, Morgan Co.

$A = .2\,p$; $B = .2\,p +$ clearance; $T = .47\,p$ (cast) and $.5\,p - \frac{1}{32}''$ (cut); clearance $= .05\,(p + 1)$ (cast), and $.03\,(p + 1)$ (cut). Angle of action $= 15°$.

See also, Stahl and Wood's, Mechanism, Page 104.

The rough cast tooth is made stronger than the machine cut tooth because at times the action of the load may be on the corner of the tooth rather than along its whole face. This may be due to the poorly shaped teeth in the pattern or to poor alignment in the machine.

125. Pitch of Gear Teeth:—The force exerted on a tooth is continually changing in value and in its line of action relative to the radius of the wheel; this makes the calculation of the gear tooth a rather unsatisfactory problem. In the following it will be assumed that the maximum load on the tooth will act at the end of the tooth and that its value will be equal to the total load on the gear. It will also be assumed that the tooth has a rectangular section throughout.

Not one of the above assumptions is exactly correct but each is accepted as being the best approximation that can be made. In explanation of the above it may be said first, that the maximum load W on the end of the tooth acts in a line that is not perpendicular to the line joining the centers of the gears although the friction between the teeth has a tendency to bring it perpendicular, how nearly this is true in practice is not an easy matter to determine, hence the maximum value is taken; second, the maximum load on the tooth is not known to be the full value of the turning force for instance, if the arc of action for a pair of gears in contact is twice the pitch there would be two pairs of teeth in contact and the pressure on one tooth would be *theoretically* $\frac{W}{2}$, but since because of poor alignment, springing shafts or irregular shaped teeth the full load may come on one tooth only it is best to design the tooth to fill such conditions; third, the tooth outline is not rectangular but varies from this on account of the *kind* and *construction* of the teeth, diameter of gear, etc. In some teeth, especially those on large wheels the outline will be much stronger while for small gears it might be much weaker than the rectangle. As an average the rectangular section is accepted.

Suppose the worst condition to be when the load W acts at the corner of the tooth Fig. 36, in which case the tooth will break along some line as $a\,d = x$ and we have

$$W\,l = \tfrac{1}{6}\,f\,x\,t^2 \qquad (1)$$

Assume both W and t constant then f is a maximum when $\frac{l}{x}$ is a maximum. Since $l = h \sin\theta$ and $x = \frac{h}{\cos\theta}$, $\frac{l}{x} = \sin\theta\cos\theta$ which is a maximum at 45° and gives $\frac{l}{x} = \frac{1}{2}$.

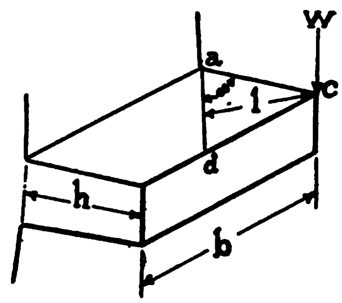


Fig. 36.

Substituting these values $l \div x = \frac{1}{2}$, and $t = .47$ p from the table, into (1) we have

$$p = 3.68\sqrt{\frac{W}{f}} \qquad (2)$$

For rough gears b = 1.5 p to 2 p.

Next, considering the force W acting along the entire face of the tooth and perpendicular to it

$$\text{W h} = \frac{f\,b\,t^2}{6}$$

from which if $t = .47$ p as before, $h = .7$ p and $b =$ n p where n = 3

$$p = 2.5 \sqrt{\frac{W}{f}} \qquad (3)$$

For machine cut teeth with $h = .6764$ p, t = .5 p, b = 3 p we have from (3)

$$.6764 \text{ p W} = \frac{3 \text{ p} f \times (.5 \text{ p})^2}{6} \qquad \text{and}$$

$$p = 2.33 \sqrt{\frac{W}{f}} \qquad (4)$$

The value of n may vary between 2 and 3. The above value is that which has been in general use. The tendency now however, is to decrease b and increase p which would necessarily reduce the value of n.

A number of formulas for the design of cut spur gears were derived by Mr. Wilfred Lewis and reported in the Proceedings of the Engineers Club of Philadelphia, Vol. X, 1893, p. 16; also the American Machinist, May 4, 1893, p. 3. These formulas may be summarized in the following:

$$p = \frac{W}{f\,b\,y} \quad \text{and} \quad p = \frac{H.\,P.}{.000007933\, D\, f\, b\, y\, (R\,P\,M)} \qquad (5)$$

where W, f p, and b are as stated above, D is the pitch diameter of the gear in inches and y is a factor depending upon the form of the tooth.

TABLE XII.

FACTOR FOR STRENGTH, y.

Number of Teeth	Involute 20° Obliquity.	Involute 15° and Cycloidal.	Radical Flanks.	Number of Teeth.	Involute 20° Obliquity.	Involute 15° and Cycloidal.	Radial Flanks.
12	.078	.067	.052	27	.111	.100	.064
13	.083	.070	.053	30	.114	.102	.065
14	.088	.072	.054	34	.118	.104	.066
15	.092	.075	055	38	.122	.107	.067
16	.094	.077	.056	43	.126	.110	.068
17	.096	.080	.057	50	.130	.112	.069
18	.098	.083	.058	60	.134	.114	.070
19	.100	.087	.059	75	.138	.116	.071
20	.102	.090	.060	100	.142	.118	072
21	.104	.092	.061	150	.146	.120	.073
23	.106	.094	.062	300	.150	.123	.074
25	.108	.097	.063	Rack.	.154	124	.075

If in (5), the length of the tooth b is not given, it may be more convenient to substitute for b, its value $n\,p$ and give a value to n.

When gears have small fillets and the number of teeth N is known

$$p = \sqrt{\frac{W\,N}{n\,f\,(0.124\,N - 0.888)}} \qquad (6)$$

In each of the above formulas f may be used according to Table XIII, which was closely verified by Prof. Benjamin after a series of experiments on the breaking of gear teeth.

Linear Velocity in ft. per min.		100 or less	200	300	500	1000	1500	2000
Cast Iron	$f =$	8000	6000	4800	4500	2500	2000	1800
Cast Steel	$f =$	20000	15000	12500	10000	7000	5000	4500

These fibre stresses will appear large when compared with those used in ordinary machine construction and should not be applied to other than gear work.

The following formula quoted by Suplee gives

$$f = \frac{9600000}{\text{F. P. M.} + 2164}$$

Application:—Given two meshing gears, with pitch circles approximately 18 inches and 6.5 inches diameter, transmitting 25 horse power at a speed of the larger gear of 150 R. P. M.; what is the pitch and the length of the teeth?

From the conditions as named V = 707 F. P. M. and W = 1167 pounds. Assuming now f = 3000 and N = 50 and substituting in the above formulas it gives for the large gear

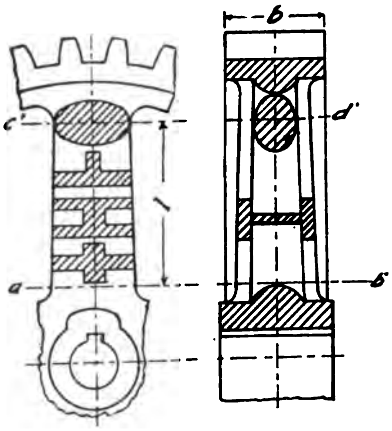
$$\begin{aligned}
&(2)\ p = 2.28''; \ b = 4.56'' \\
&(3)\ p = 1.55''; \ b = 4.65'' \\
&(4)\ p = 1.44''; \ b = 4.32'' \\
&(5)\ p = 1.08''; \ b = 3.24'' \\
&(6)\ p = 1.12''; \ b = 3.36''
\end{aligned}$$

and for the small gear (2), (3) and (4), p is the same as above.

$$\begin{aligned}
&(5)\ p = 1.28''; \ b = 3.84'' \\
&(6)\ p = 1.45''; \ b = 4.35''
\end{aligned}$$

It is evident from this that (4) and (6) give the same results on pinions of 16 teeth but on larger gears (6) gives a smaller value than (4). In any case however, where two gears mesh together both pitches must be the same as that required for the smaller of the two, hence in the above application the pitch of the large gear would not be considered.

126. Size of Gear Arms:—The size of the arm at the center of the wheel may be figured by moments, where $f = 2500\#\square''$; the load on the arm = $W \div$ No. of arms; and l = radius of gear. Having the section at the center, that at the rim is given the usual proportion.



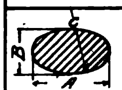
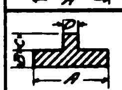
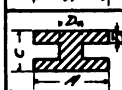
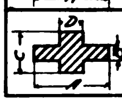
$b = 3.5p - \sqrt{\frac{p}{2}}$		p = Pitch			
		A	B	C	D
	Sec a'b'	2p+.06l	p+.03l	$\frac{3}{4}$A	
	Sec. c'd'	2p	p	$\frac{3}{4}$A	
	Sec. a'b'	2.25p+.06l	.5p	b-1.6B	.3p
	Sec c'd'	2.25p	.5p	b-2B	.3p
	Sec a'b'	2.1p+.06l	.5p	b-.5p	.3p
	Sec c'd'	2.1p	.5p	b-.5p	.3p
	Sec a'b'	2.3p+.06l	.5p	b-1.5B	.3p
	Sec. c'd'	2.3p	.5p	b-2B	.3p

FIG. 37.

Fig. 37 gives approximate sizes for sections that are sometimes recommended.

127. Bevel Gears:—To determine the pitch diameter, pitch, and number and length of teeth of bevel gears the following based upon the formula of spur gears by Wilfred Lewis, gives fairly good results.

In Fig. 38 A, let l = slant height of cone of revolution
b' = length of bevel gear tooth taken where possible at $l \div 3$
b = length of equivalent spur gear tooth
$r = b' \div l$
then from American Machinist, May 31, 1900,

$$b' = \frac{b}{1 - r + (\frac{1}{3}r^2)} \qquad (7)$$

Formula (7) is derived upon the assumption that b Fig. 96 A is the breadth of a spur gear of the same number of teeth and whose pitch diameter is equal to the largest pitch diameter of the cone of revolution on the bevel gear, hence to plan the bevel gear assume a diameter and find b and p of the spur gear, after which substitute b and r in formula (7) for b'

If after substituting the value r, say .33, and $b \div l$ is found to differ much from the assumed value of r it will be necessary to assume another value of r or a new diameter for the spur gear and solve again.

Application:—Required a set of mitre gears of approximately 18 inches pitch diameter to transmit 25 horse power at 150 R. P. M.

From the application under spur gears p = 1.12 inches and b = 3.36 inches; substituting the value b in equation (7) it gives

$$b' = \frac{3.36}{1 - \frac{1}{3} + \frac{1}{27}} = 4.7 \text{ inches}$$

In a mitre gear as above, $l = 12.75$ inches and $4.7 \div 12.75 = .37$ instead of .33 as first assumed. Now if r be taken at .4 instead of .33 it gives

$$b' = \frac{3.36}{1 - \frac{2}{5} + \frac{4}{75}} = 5.1 \text{ inches}$$

and $5.1 \div 12.75 = .4$ which checks exactly. The value of $r = .33$ could have been maintained by assuming a diameter slightly larger than 18 inches and solving again.

The above method works fairly well when the gears are mitre and of moderate size. It does not work so well, however, for pinions and gears differing widely in angular velocity. A formula which may be more generally applied is the orignal one by Mr. Lewis,

$$W = f\,p\,b'\,y\,\frac{D^3 - d^3}{3\,D^2\,(D - d)} \tag{8}$$

where f, and p are the same as given in (5); W is the load in pounds on the largest pitch cone; D and d are the largest and smallest diameters, respectively, of the frustrum of the cone; b' is the length of the bevel gear tooth, and y is taken from table corresponding to *formative number of teeth, i. e.* Actual number of teeth $\times$ secant a. Where a = slope of cone.

In most cases d will not be less than $2\,D \div 3$ and (8) will give good results if taken

$$W = f\,p\,b'\,y\,\frac{d}{D} \tag{9}$$

Note.—In applying (8) or (9), it will be found more convenient to assume the gear sizes and solve for the force W. If, however, W is given to find the size of the gear, proceed as follows: find p as in a spur gear; take f and y from the tables; assume D, and by construction find a relation between d and b'; then by substitution either d or b' may be obtained.

Another method which will give fairly satisfactory results, is to make the pitch and the length of the bevel gear tooth, equal to those of a spur gear, whose pitch diameter is equal to the mean pitch diameter of the frustrum of the cone of the bevel wheel.

In planning for bevel gears it is customary to locate them either ***against*** or ***near to*** a journal box. It will be seen, B, Fig. 38, that in addition to the torsion on the shaft due to the force P, = W, there is a bending of the shaft due to the force s which is a resultant of the force m tending to separate the gears. It is therefore evident that the nearer the gear is to the support, the less the bending of the shaft.

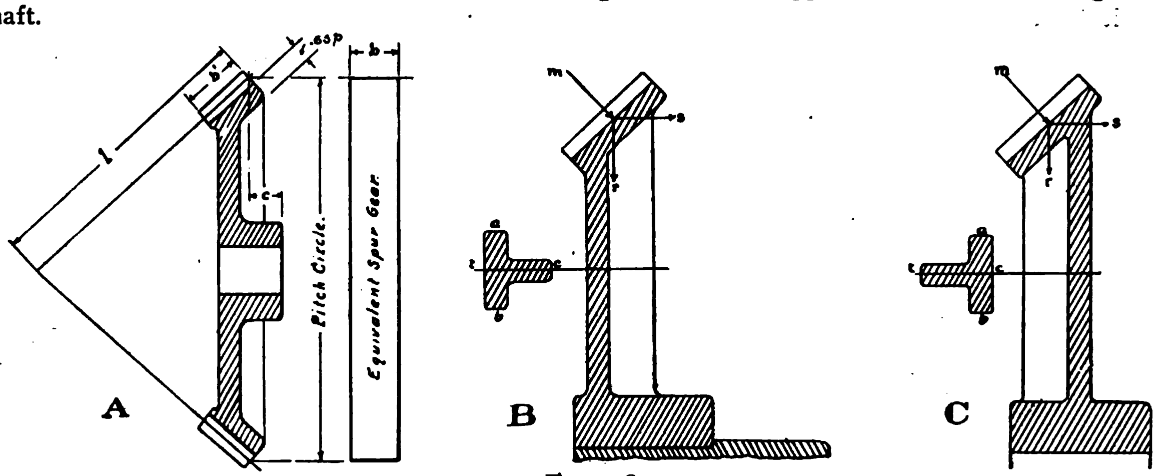


FIG. 38.

The rims of small gears are connected to the hubs by a solid disc as in A. In larger gears arms are made as in B and C. The T shaped arm is very common in gear work. The disc part of the arm ab is figured to stand the entire turning moment and the web is added to resist the side thrust s. B is stronger to resist the bending than C, but the latter is more commonly used because of its smooth back which adds safety in operation. It is undoubtedly true also that the rim in C is better supported than in B.

In fitting a shaft to a bevel gear it is usually ***shouldered*** as shown in B.

The thickness of the rim under the tooth may be taken .65 p and the backing c may be taken for ordinary conditions at

$$c = \begin{cases} \frac{1}{16}\,D + \frac{1}{4}'' \text{ for gears below } 24'' \text{ diameter.} \\ \frac{1}{8}\,D - 1'' \text{ for gears above } 24'' \text{ diameter.} \end{cases}$$

128. Friction Gearing:—Friction wheels are used in light power work where the service is intermittent, where the velocity ratio of the wheels is changeable, and where high speeds would cause toothed wheels to be noisy. They are generally used to connect shafts that are parallel or at 90 degrees with each other, but may be used to connect shafts at any angle.

The *materials* used in friction wheels are iron, wood, paper or mill-board, and leather. The driver should be made from the softer material and the follower from the harder. This is a protection against wearing the face of the follower unevenly in case of slipping. The usual materials employed are, paper for the driver and iron for the follower.

The relation existing between the pressure on the wheels perpendicular to the surface at the line of contact and the rotative force transmitted may be shown for the different conditions as follows:

For *parallel shafts* Fig. 39, let W = pressure in pounds total, w = pressure in pounds per lineal inch of wheel face b, P = total rotative force in pounds and Φ = coefficient of friction between the two surfaces; then $W\,\Phi = P$, or $b\,w\,\Phi = P$. Substituting these values in the standard horse power formula we have, if V = velocity of the rim in F. P. M.,

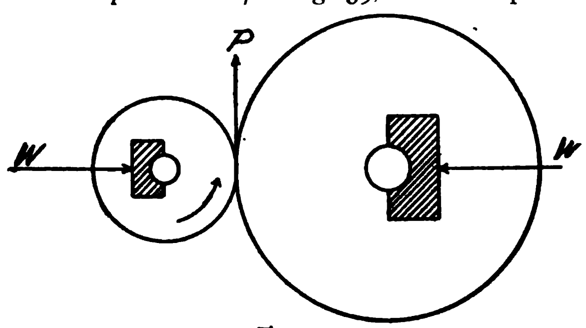


FIG. 39.

$$H.\,P. = \frac{W\,\Phi\,V}{33000} = \frac{b\,w\,\Phi\,V}{33000}. \qquad (1)$$

This formula may be used in determining the power transmitted by any given set of wheels, or to determine the required pressures between the surfaces of any two given wheels transmitting a given power. The coefficient of friction Φ may be taken according to Unwin, page 283, as follows: Metal to metal, .15 to .2; paper to metal .2; and wood to metal .25 to .3.

For *shafts at 90 degrees* Fig. 40, let W be the pressure normal to the surface as before. Resolve this into forces R_1, S_1, R_2 and S_2, parallel and perpendicular to the respective shafts. Then

$$R_1 = W \sin a = S_2$$
$$R_2 = W \cos. a = S_1$$

substituting these in the standard horse power formula we have :

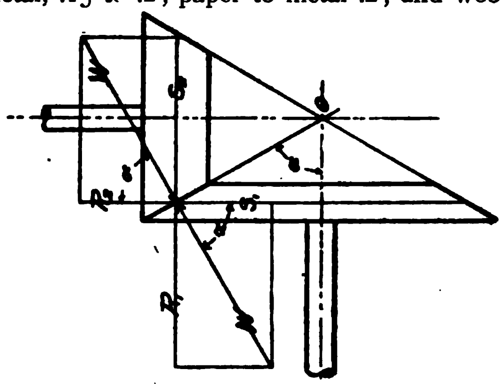


FIG. 40.

$$\text{H. P.} = \begin{cases} \dfrac{R_1\,\Phi\,V}{33000 \sin a} = \dfrac{S_2\,\Phi\,V}{33000 \sin a} \\[2ex] \dfrac{R_2\,\Phi\,V}{33000 \cos a} = \dfrac{S_1\,\Phi\,V}{33000 \cos a} \end{cases} \qquad (2)$$

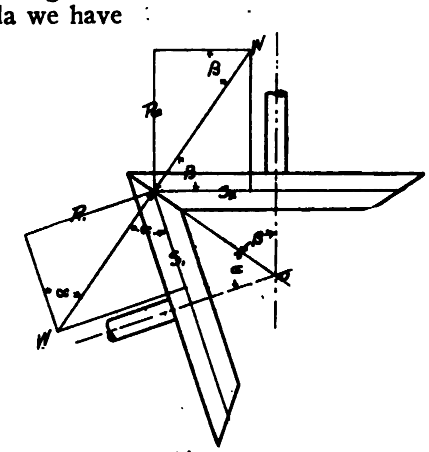


FIG. 41.

For *shafts at any angle* Fig. 41, let the notation be as before and the following formulas will be true:

$$R_1 = W \sin a.$$
$$S_1 = W \cos a.$$
$$R_2 = W \sin B.$$
$$S_2 = W \cos B.$$

then by substitution

$$\text{H. P.} = \frac{R_1\,\Phi\,V}{33000 \sin a} = \frac{S_1\,\Phi\,V}{33000 \cos a} = \frac{R_2\,\Phi\,V}{33000 \sin B} = \frac{S_2\,\Phi\,V}{33000 \cos B} \qquad (3)$$

129. To determine the width of the face of a friction wheel, use the latter part of formula (1). Unwin quotes the values $w\ \Phi$ as, 30 pounds for maple wood, 15 to 20 pounds for pine wood and 80 pounds for paper. The statement is also made that the width of face of the friction wheel may be taken the same as the width of a single leather belt transmitting the same power at the same velocity.

In the use of friction wheels, two factors enter which are more or less uncertain and difficult to determine. One, the coefficient of friction, has just been mentioned, and the other is the slippage between the two wheels. Probably no values may be quoted for them with more assurance, than those relating to paper and iron contact. A series of experiments conducted in the laboratory of Purdue University, and summarized in a paper by Dr. W. F. M. Goss before the A. S. M. E. December, 1896, give results that are quoted as standard authority. The experiments involved paper friction wheels of 5½, 8, 12 and 16 inches diameter, in contact with a 16 inch cast iron wheel. The contact pressure varied from 75 pounds per inch of width to more than 400 pounds, and the speed limits gave a peripheral velocity varying from 450 to 2700 feet per minute. More than 5000 observations were made. The following is a summary of the results:

130. *"Slippage.*—By increasing the load to be carried, the slippage may always be gradually increased to three per cent., and under favorable conditions its gradual increase may reach a maximum of six per cent., but when the slippage is between the limits of three per cent. and six per cent. it is likely to undergo a rapid increase to 100 per cent.; that is, the driven wheel is likely to stop."

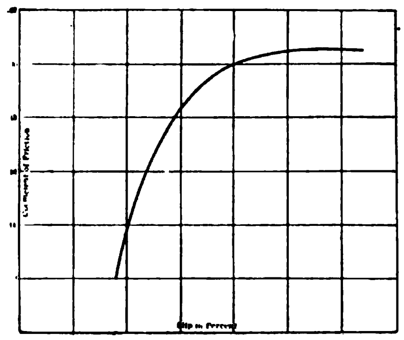


FIG. 42.

131. *"The Coefficient of Friction* depends upon conditions some of which have not been studied and are not understood. It is most affected by slippage. Its value increases with increase of slip until the latter becomes about three per cent, after which the action of the gearing becomes uncertain. With a slippage of two per cent., the maximum value of the coefficient rises above twenty-five per cent., and as the slippage approaches three per cent., even larger values have been observed. Fig. 42 shows a relation between slippage and the coefficient of friction, which can easily be maintained with paper friction wheels of 8 inches or more in diameter.

The coefficient of friction is apparently constant for all pressures of contact up to a limit which lies between 150 and 200 pounds per inch of width of wheel-face, beyond which limit its value decreases. At 400 pounds pressure its value is from ten per cent. to fifteen per cent. less than for 150 pounds.

Friction wheels of 8, 12, and 16 inches diameter give nearly the same value for the coefficient, while results from a six inch wheel are lower by about ten per cent.—a fact which would seem to indicate that wheels smaller than those experimented upon may have a still lower value for their coefficient.

Variations in peripheral speed between 400 and 2,800 feet per minute do not affect the coefficient of friction."

132. *"Pressure of Contact.*—With a constant coefficient of friction, the power transmitted varies directly with the pressure of contact. During the comparatively short period covered by the experiments the paper wheels gave no indications of breaking down under pressure as high as 400 pounds per inch in width. The work was not continued through a period sufficiently long, however, to permit a determination of the maximum pressure with which paper drivers may be forced against their iron followers; but it has already been noted that the coefficient of friction is maximum under a pressure of about 150 pounds per inch in width, and while the amount of power delivered may be augmented by increasing the pressure above this limit, the most efficient pressure is that for which the coefficient of friction is maximum."

133. *"Horse Power.*—By making d the diameter of the friction wheel in inches, w the width of its face also in inches, and N the revolutions per minute, and by accepting 0.2 as a safe value for the coefficient of friction, and a pressure of 150 pounds per inch width of face as the pressure of contact, the horse-power may be written as,

$$\text{H. P.} = \frac{150 \times 0.2 \times \pi\, d \times w \times N}{12 \times 33000} = .000238\, d\, w\, N.$$

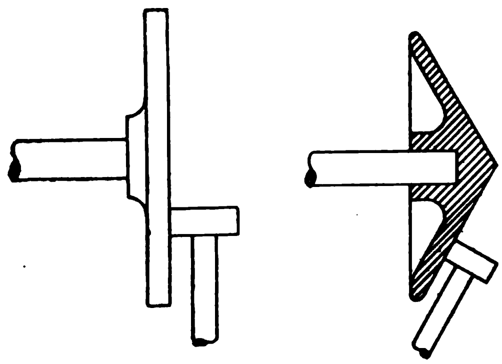
This formula is believed to be safe for friction wheels which are 8 inches or more in diameter, and under conditions which make it possible for them to be kept reasonably clean. By its use the following table has been calculated:"

TABLE XIV.

HORSE-POWER WHICH MAY BE TRANSMITTED BY MEANS OF A CLEAN PAPER FRICTION WHEEL OF ONE INCH FACE WHEN RUN UNDER A PRESSURE OF 150 POUNDS.

Diameter of Pulley.	Revolutions Per Minute.										
	25	50	75	100	150	200	300	400	600	800	1000
4	.0238	.0476	0714	.0952	.1428	.1904	.2856	.3808	.5712	.7616	.9520
6	.0357	.0714	.1071	.1428	.2142	.2856	.4284	.5712	.8568	1.1424	1.428
8	.0476	.0952	.1428	.1904	.2856	3808	.5712	.7616	1.1424	1.5232	1.904
10	.0595	.1190	.1785	.2380	.3570	.4760	7140	.9520	1.4280	1 9040	2.380
14	.0833	.1666	.2499	.3332	.4998	.6664	.9996	1.3328	1 9992	2.6656	3.332
16	.0952	.1904	.2856	.3808	5712	.7616	1.1424	1.5232	2.2848	3.0464	3.808
18	.1071	.2142	.3218	.4284	.6426	.8568	1.2852	1.7136	2.5704	3.4272	4.284
24	.1428	.2856	4284	.5712	.8568	1 1424	1 7136	2.2848	3.4272	4.5696	5.712
30	.1785	.3570	.5355	.7140	1 0710	1.4280	2.1420	2.8560	4 2840	5.7120	7.140
36	.2142	.4284	.6426	.8568	1.2852	1.7136	2 5704	3.4272	5.1408	6.8544	8.568
42	.2499	.4998	.7497	9996	1 4994	1.9992	2 9988	3.9984	5.9976	7.9968	9.996
48	.2856	.5712	.8568	1.1424	1 7136	2.2848	3 4272	4.5696	6.8544	9.1392	11 '24

134. Another form of friction gearing is the *disc friction* shown in Fig. 43 *A*, for shafts at 90 de grees, or the *cone friction* shown in *B*, for shafts at any angle. These are used to obtain variable speeds It is evident that if the rotative speed of the driver be constant, the corresponding speed of the follower will depend upon the position of the driver relative to the center of the follower. Such gears are used on sensitive drills and other similar light machinery. In disc or cone frictions, a certain slippage takes place because it is not a pure rolling contact. This slippage can not be entirely eliminated.



A B

FIG. 43.

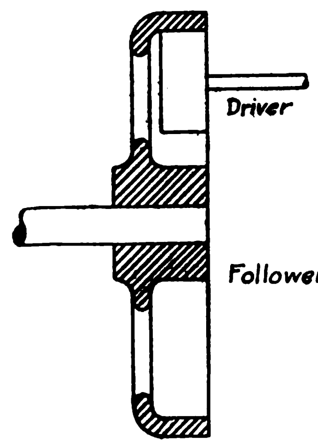
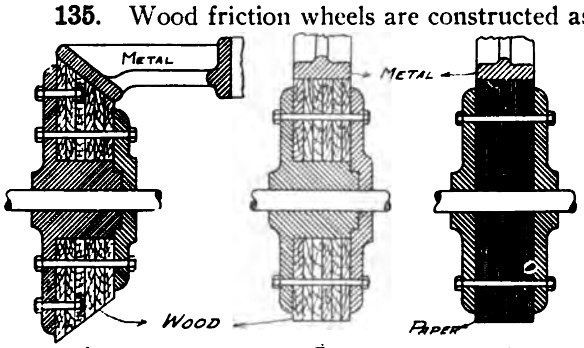


FIG. 44.

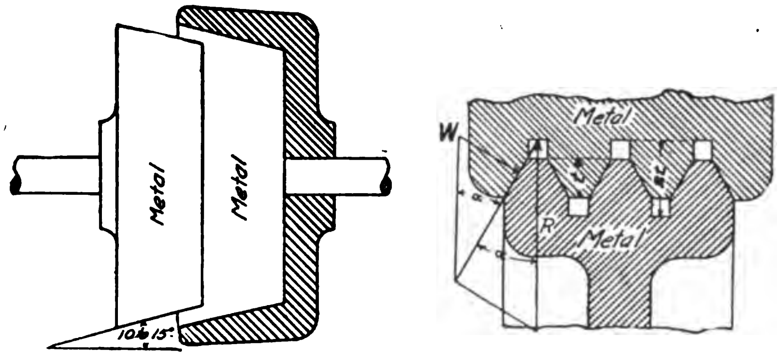
Fig. 44 shows a *reversible friction* which may be used on machines requiring slow forward and fast returning movements. With a constant speed of the driver, a slow forward speed may be obtained by forcing the driver against the inner surface of the outside rim, and to reverse, the driver may be forced against the outside surface of the hub of the wheel.

135. Wood friction wheels are constructed as shown in *A* and *B*, Fig. 45. Paper friction wheels are constructed as in *C*. The wood should be very uniform in texture, free from knots, and fine grained. Clear maple, pine or cottonwood may be used. The paper generally used is straw-board cut in large discs to fit closely to the shaft. These when compressed tightly between the flanges make a very dense and uniform material. In all flanges the bolt heads and nuts should be protected by ribs, or should be let into the flanges.



A B C

FIG. 45.



A B

Fig. 46.

136. Wedge Frictions are of two general classes, Fig. 46 *A*, two shafts in the same straight line, and *B*, two shafts in parallel. The first is used generally on power hammers and the second is used only on slow speed machinery because of the noise attending its use. The number of projections on *B* is between 1 and 6. Unwin gives the relation between *R*, the force holding the wheels together, and *P* the turning force on the shaft as

$$R = \frac{P}{\Phi} (\sin a + \Phi \cos a)$$

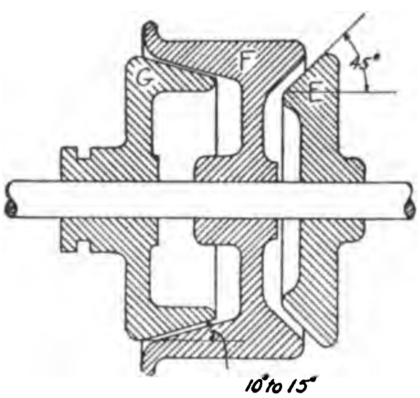


Fig. 47.

where Φ is the coefficient of friction and a is the angle of inclination between the side of the wedge and the vertical. This is usually taken between 15 and 20 degrees. The working depth of the tooth is

$$t = 0.025 \sqrt{P}$$

Fig. 47, shows a very effective form of friction gearing. In this, *E* is keyed fast to the shaft; *G*, fits over a sliding key and *F* is loose. When *G* is thrown to the right it wedges *F* between *G* and *E* and causes rotation of the shaft. This form gives a more positive action than the single frictions previously mentioned.

WORM GEARING.

137. Worm Gearing:—In worm gearing Fig. 48, the worm is merely a long tooth wrapped around a cylinder; the pitch of the worm being the same as the pitch of the gear. This gearing is used in changing from a very high to a very low speed and is employed in the transmission of only small powers. The worm is the driver and, unless the pitch of the thread is very great, it will effectually lock the gearing. The wear on the worm is much greater than that of the gear.

The action of the worm in connection with the gear is very much the same as a rack and spur gear. One revolution of the worm moves the gear through one circular pitch. The *ratio* of the *angular velocities* of the two shafts is the same as the *number of teeth on the gear*. It is advisable to make this ratio as large as possible. The minimum value is sometimes quoted as 30. When a smaller ratio is desired a *double thread* on the worm may be employed. A ratio very often used is 100. This value may be considered standard.

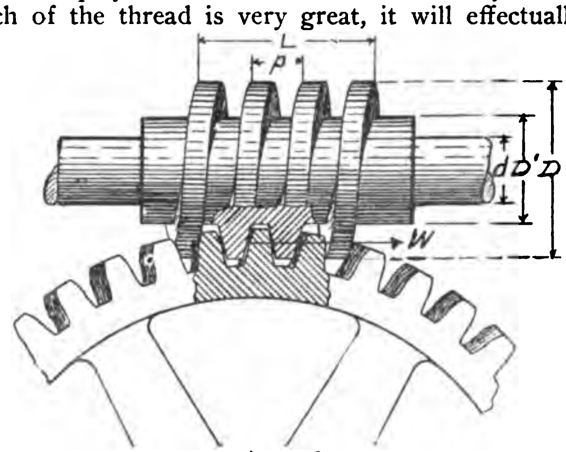


Fig. 48.

In designing the worm the following values will be found very good. Take the pitch p the same as that of a spur gear having the same W, then

$D = 4\ P$

$D' = 1.8\ d$ to $2\ d.$

$L = 3\ p$ to $6\ p$, say $4\ p$.

The teeth of the wheel may be straight and set at an angle from the center line of the shaft so as to conform with the slant of the worm thread, or they may be curved to fit the worm thread along the full length of the gear tooth. The latter method is preferred since, in the former, there is contact only along the center line of the gear.

Fig. 49 shows sections through the gear rim. *A*, is that of a straight tooth; *B*, is the ordinary form used on curved teeth, and *C*, is a modification of the curved tooth, having its upper face concentric with the outside of the worm hub.

The tooth may be given the usual values for addendum, dedendum, thickness, clearance, etc. The width of the face of the gear varies from 1.5 *p to* 2.5 *p*. The involute tooth is preferred, because the tool which cuts the worm is a simpler form.

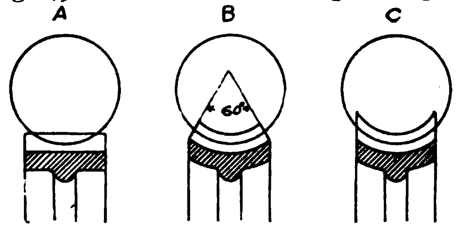


FIG. 49.

Right and left hand worms, Fig. 50, may be upon the same shaft, meshing into two gears that run in opposite directions. These gears operate two shafts, which in turn are connected by spur gears. The one shaft is an idler shaft, but serves to minimize the end thrust on the worm shaft.

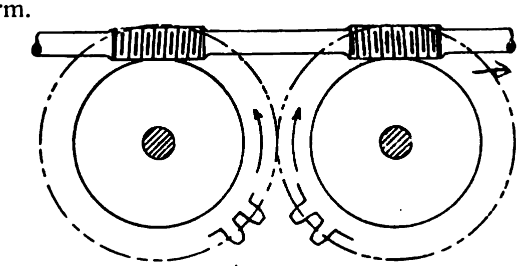


FIG. 50.

SCREW GEARING.

138. Screw Gearing:—Most fastenings between machine parts are made through the action of the screw thread. The threads are classified as *square* and *V*, and the pieces thus threaded are classified as *bolts, studs, cap screws, set screws* and *machine screws*. See Par. 142., for standard sizes.

The *action of the loaded screw,* when it is turned about its own center, is equivalent to moving the same load along an inclined plane whose angle with the horizontal is the same as the mean pitch angle of the thread.

139. *Work Performed by Means of a Screw*:—A very simple illustration of the overcoming of resistance by means of a screw is shown by Figure 51.

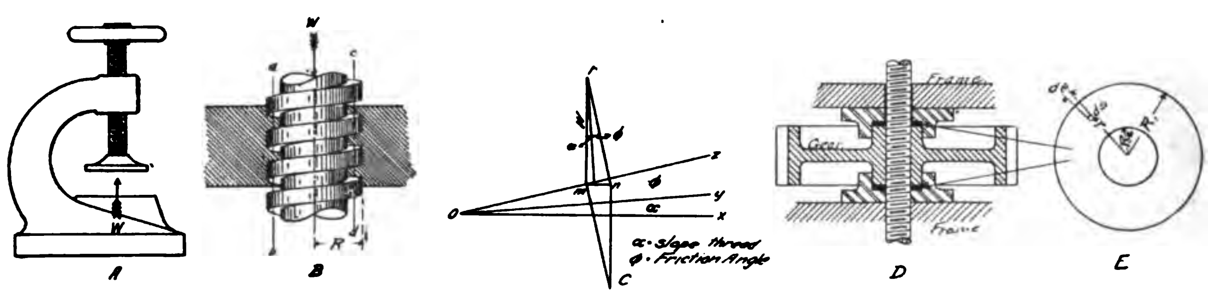


Fig. 51.

In *A* the square thread screw is sustaining a load of *W* pounds. Let this load be absorbed on a circle represented by the mean circumference of the thread whose diameter lies between *ab* and *cd* in *B*. Lay off this mean circumference *ox* in *C* and find the slope of the screw *oy*. Assume a coefficient of friction between the screw and the nut and lay off the line *oz* an angle Φ from *oy*.

With the force *W* perpendicular to *ox*, draw *rn* making the angle $(a + \Phi)$ with *W*. Draw *mn* perpendicular to *W*, *mn* represents the turning effort *P'* in pounds, necessary to be applied at a radius *R'* from the center of the screw to move the load up the inclined plane against the action of its own

weight and friction. One complete turn of the screw is equivalent, in effective work, to moving the load through the distance of one pitch.

Application. 1. The main screw on a tension testing machine acts directly upon the test specimen. The maximum pull of the specimen is estimated at 15000 pounds. The screw is 2 pitch, is 2 inches mean diameter and is operated by a gear having 12 inches pitch diameter. What force will be exerted between the gear and the pinion in overcoming the load and the friction of the screw, if the latter coefficient be taken at .15?

The development of the screw thread gives an angle of $4° — 30$ and the angle represented by the coefficient of friction is $8° — 30'$ making the total angle of $13°$. From this is obtained

$$\tan. 13° = mn \div W \text{ and}$$
$$p' = mn = 3463 \text{ pounds.}$$

From moments we can readily obtain

$$3463 \times 1 = PR$$
$$\text{where } R = \text{radius of gear} = 6''$$
$$\text{and } P = 3463 \div 6 = 577 \text{ pounds.}$$

If instead of the gear, a two ended lever be used, the force exerted on each end of the lever would be $577 \div 2 = 288.5$ pounds.

Application 2: Given, the main screw of the above with the gear threaded to fit and with the end of the hub encased in an annular thrust bearing having 2.25 inches and 4.5 inches respectively, as the inner and outer diameters, Fig. 51 D. What would be the total force P on the gear necessary to overcome the load, the friction of the screw and the friction of the thrust bearing?

Considering first the thrust bearing, the turning moment necessary to overcome this friction is

$$P R = \tfrac{2}{3} \pi \Phi p (R_1^3 - R_2^3) \qquad (1)$$

where P and R are the same as previously stated; Φ = coefficient of friction = say .15; p = pressure per square inch of area on thrust bearing, and R_1 and R_2 are the radii respectively of the outer and the inner circumferences of the bearing.

Formula (1) represents the *frictional resistance offered by any annular disc. Proof.* Let $d\ s$ Fig. 51 E, be any increment of area at any radius r from the center of the disc. Then $d\ s = d\ r\ r\ d\ \theta$ and $r\ d\ s = r^2\ d\ r\ d\ \theta$. From this is obtained

$$P R = \int_{R_2}^{R_1} \int_{o}^{2\pi} p\ \Phi\ r^2\ d\ r\ d\ \theta \text{ and}$$

$$P R = \tfrac{2}{3} \pi \Phi\ \ p\ [R_1^3 - R_2^3], \text{ as given in} \qquad (1)$$

Substituting the above conditions in (1) we obtain

$$6 P = \frac{2 \times 3.14 \times .15 \times 1258}{3} [(2.25)^3 - (1.125)^3]$$

$$P = 656.9 \text{ pounds on the thrust bearing.}$$

Then $577 + 656.9 = 1233.9$ pounds, total force at gear for load and frictions.

For the two ended lever this would be 616.9 pounds on each end of the lever.

140. The *energy* exerted in one complete revolution of the screw is (not counting the thrust bearing) $577\ \pi\ D = 1813.4$ foot pounds; and counting the friction of the thrust bearing it is $1233.9\ \pi\ D = 3873.$ foot pounds.

The *useful work* done is $15000 \times .5 = 7500$ inch pounds or 625 foot pounds for each revolution.

The *efficiency* of the screw (including the thrust bearing) is $625 \div 3878 = .16$; and without the thrust bearing it is .34. This latter efficiency may also be shown as $\text{Tan}\ a \div \text{Tan}\ (a + \Phi)$.

Where screws are used for transmission purposes, the square thread is preferred. This choice is made largely because of the *wedging action* of the V thread, thus having a tendency toward splitting the nut. This wedging force on a V thread may be obtained in any case if desired. Let $W =$

force parallel with the center of the screw, a = the angle of the screw thread, and Q = force perpendicular to W; then $Q = W \tan (a \div 2)$. Since the standard American screw thread is 60 degrees, this formula reduces to

$$Q = .577\,W. \qquad (2)$$

141. To obtain the *force necessary to shear* or strip a V thread from the bolt, multiply the circumference of the bolt at the root of the thread by the length of the nut and by the fibre stress of shearing. To obtain the corresponding force to strip the thread from the nut, multiply the circumference of the outside of the thread by the length of the nut and by the fibre stress of shearing. For an ordinary bolt and nut, the stripping takes place on the threads of the bolt. The length of the thread in action at any time should be such that the force necessary to strip the thread will be greater than the tensional strength of the bolt at the root of the thread.

The minimum calculated length for a 1 inch wrought iron bolt in a wrought iron nut is approximately .3 inch This, in the table, is quoted as 1 inch. The minimum calculated length of thread in the cast iron nut or casing, if used, is approximately .75 inch. This in practice is taken at about 1.5 × diameter of bolt. From this it is seen that the acting length of the screw thread is made in practice from two to three times the calculated length.

If two castings be bolted together, and one be tapped for a screw fastening, a cap screw may be used where the pieces will seldom need separation. If, however, the pieces frequently need to be taken apart, the stud bolt should be used. The brittleness of the cast iron thread renders it more susceptible to wear, hence the above statement.

142. The *forms of the Standard threads* are shown in Fig. 52.

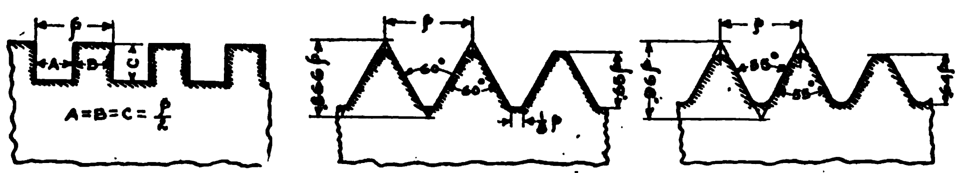
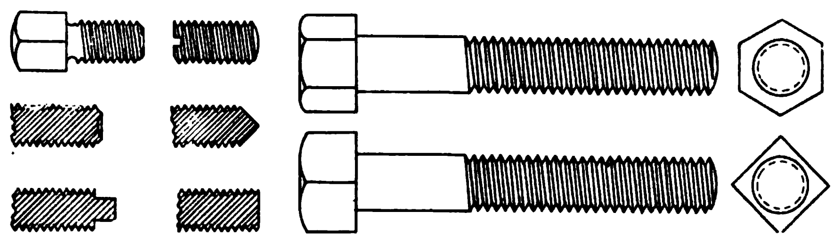


Fig. 52.

The following tables No. 15, refer to the United States standard or Sellers threads.



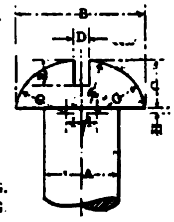
SET SCREWS			HEX.HEAD CAP-SCREWS.			SQ. HEAD CAP-SCREWS.		
Short Diam. of Head.	Long Diam of Head	Lengths (under Head)	Short Diam. of Head.	Long Diam of Head	Lengths (under Head)	Short Diam. of Head.	Long Diam of Head	Lengths (under Head)
I/4	.35	3/4 - 3	7/I6	.5I	3/4 - 3	3/8	.53	3/4 - 3
5/I6	.44	3/4 - 3 I/4	I/2	.58	3/4 - 3 I/4	7/I6	.62	3/4 - 3 I/4
3/8	.53	3/4 - 3 I/2	9/I6	.65	3/4 - 3 I/2	I/2	.7I	3/4 - 3 I/2
7/I6	.62	3/4 - 3 3/4	5/8	.72	3/4 - 3 3/4	9/I6	.80	3/4 - .3 3/4
I/2	.7I	3/4 - 4	3/4	.87	3/4 - 4	5/8	.89	3/4 - 4
9/I6	.80	3/4 - 4 I/4	I3/I6	.94	3/4 - 4 I/4	II/I6	.98	3/4 - 4 I/4
5/8	.89	3/4 - 4 I/2	7/8	I.0I	I - 4 I/2	3/4	I.06	I - 4 I/2
3/4	I.06	I - 4 3/4	I	I.I5	I I/4 - 4 3/4	7/8	I.24	I I/4 - 4 3/4
7/8	I.24	I I/4 - 5	I I/8	I.30	I I/2 - 5	I I/8	I.60	I I/2 - 5
I	I.42	I I/2 - 5	I I/4	I.45	I 3/4 - 5	I I/4	I.77	I 3/4 - 5
I I/8	I.60	I 3/4 - 5	I 3/8	I.59	2 - 5	I 3/8	I.95	2 - 5
I I/4	I.77	2 - 5	I I/2	I.73	2 - 5	I I/2	2.13	2 I/4 - 5

NOTE.- The nominal DIAMETER OF SCREW in each of the above is the same as the first column under "Set Screws".

The following tables, No. 16, referring to the sizes of machine screws, were proposed by a committee of the A. S. M. E. and were acted upon at the May Meeting, 1906.

TABLES XVI. MACHINE SCREWS.

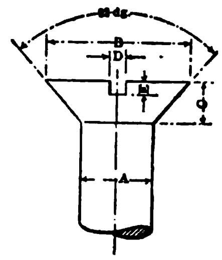
Round Head Machine Screws.



A = Diameter of Body.
$B = 1.83A$ = Diameter of Head.
$C = 0.703A$ = Thickness of Head.
$D = 0.235A$ = Width of Slot.
$E = 0.40A$ = Depth of Slot.
$F = 1.095A$ = Rad. of Top of Head.
$G = 0.70A$ = Rad. of Sides of Head.
$H = 0.068A$ = Dist. from bottom of Head to center of G.
$I = 0.213A$ = Dist. from center of Head to center of G.

A	B	C	D	E	F	G	H	I	Thread per In
.070	1281	0492	0164	0280	..0766	0490	0048	.0149	72
.085	1555	0597	0200	0340	0931	0595	0058	0181	64
.100	1830	0703	0235	0400	1095	0700	0068	0213	56
.110	.2013	0773	0258	0440	1204	0770	0075	0234	48
.125	.2287	.0879	0294	0500	1369	0875	0085	0266	44
.140	.2562	0984	.0329	0560	1533	0980	0095	0298	40
.165	3019	.1160	.0388	.0660	1807	1155	0112	.0351	36
.190	3477	.1336	0446	0760	2080	1330	0129	.0405	32
.215	.3934	1511	0505	0860	2354	1505	0146	.0458	28
.240	4392	1687	0564	0960	2628	1680	.0163	.0511	24
.250	4575	1757	0587	1000	2737	.1750	.0170	.0532	24
.270	4944	1898	0634	1080	2956	.1890	.0184	.0575	22
.320	5856	2250	0752	1280	.3504	2240	0218	.0682	20
.375	6862	2636	0881	1500	4106	.2625	0255	.0799	16

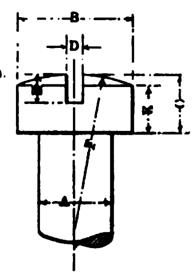
Flat Head Screws.



A = Diameter of Body.
$B = 2A - 0.0052$ = Diameter of Head.
$C = \frac{A - 0.0052}{1.739}$ = Thickness of Head.
$D = 0.235A$ = Width of Slot.
$E = \frac{1}{3}C = \frac{A - 0.0052}{5.217}$ = Depth of Slot.

A	B	C	D	E	Threads per Inch
070	1348	0373	0164	0124	72
085	1648	0459	0200	0153	64
100	1948	0545	0235	0182	56
110	2148	0603	0258	0201	48
125	2448	0689	0294	0229	44
140	2748	0775	0329	0258	40
165	3248	0919	0388	0306	36
190	3748	1063	0446	0354	32
215	4248	1206	0505	0402	28
240	4748	1350	0564	0450	24
250	4948	1408	0587	0469	24
270	5348	1523	0634	0508	22
.320	6348	1816	0752	0605	20
375	7448	2126	0881	0709	16

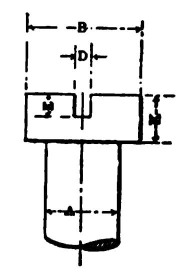
Oval Fillister Head Screws.



A = Diameter of Body.
$B = 1.6A$ = Diameter of Head.
$C = 0.80A$ = Thickness of Head (oval).
$D = 0.235A$ = Width of Slot.
$E = 0.5C = 0.4A$ = Depth of Slot.
$F = 2.186A$ = Radius of Head.
$K = 0.65A$ = Thickness of Head (flat).

A	B	C	D	E	F	K	Threads per In.
.070	1120	0560	0164	0280	1530	0455	72
.085	1360	0680	0200	0340	1858	.0552	64
100	1600	0800	0235	0400	2186	.0650	56
110	1760	0880	0258	0440	2404	.0715	48
125	2000	1000	0294	0500	2732	.0812	44
140	2240	.1120	0329	.0560	3060	0910	40
165	2640	.1320	0388	0660	3607	.1072	36
190	3040	.1520	0446	.0760	4153	.1235	32
215	3440	.1720	0505	.0860	4700	1397	28
240	3840	.1920	.0564	.0960	5246	1560	24
250	4000	2000	0587	.1000	5465	1625	24
270	4320	.2160	0634	.1080	5902	.1755	22
320	5120	.2560	0752	.1280	6995	.2080	20
375	6000	3000	0881	.1500	8197	.2437	16

Flat Fillister Head Screws.

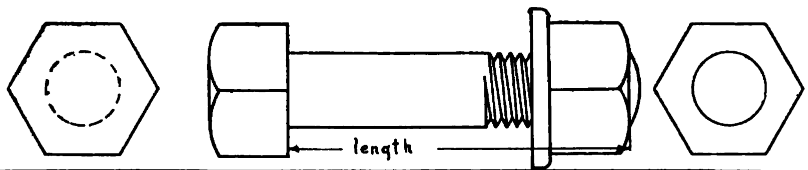


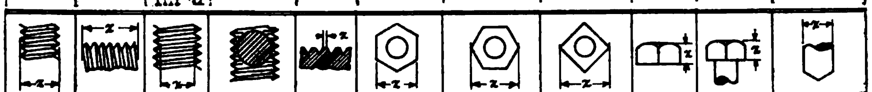
A = Diameter of Body.
$B = 1.6A$ = Diameter of Head.
$K = 0.65A$ = Thickness of Head.
$D = 0.235A$ = Width of Slot.
$E = 0.5K = 0.325A$ = Depth of Slot.

A	B	K	D	E	Threads per Inch.
.070	1120	0455	0164	0227	72
.085	1360	0552	0200	0276	64
.100	1600	0650	0235	0325	56
.110	1760	0715	0258	0357	48
.125	2000	0812	0294	0406	44
.140	2240	0910	0329	0455	40
.165	2640	1072	0388	0536	36
.190	3040	1235	0446	0617	32
.215	3440	1397	0505	0698	28
.240	3840	1560	0564	0780	24
.250	4000	1625	0587	0812	24
270	4320	1755	0634	0877	22
.320	5120	2080	0752	1040	20
.375	6000	2437	0881	1218	16

Standard Sizes of Machine Bolts and Nuts:

TABLE 17.





Diam. of screw	Thread per Inch.	Diam of Root of Thr'd	Area of Root of Thread	Width of Flat	Short Dia. of Hexagon	Long dia. of Hexagon	Long dia Square	Thickness of Nuts	Thickness of Heads	Tap Drill
1/4	20	.185	.02688	.0062	1/2	37/64	7/10	1/4	1/4	3/16
5/16	18	.24	.045238	.0074	19/32	11/16	10/12	5/16	19/64	1/4
3/8	16	.294	.067886	.0078	11/16	31/64	63/64	3/8	11/32	5/16
7/16	14	.344	.092940	.0089	25/32	29/32	1 7/64	7/16	25/64	23/64
1/2	13	.4	.125664	.0096	7/8	1	1 15/64	1/2	7/16	13/32
9/16	12	.454	.183984	.0104	31/32	1 1/8	1 23/64	9/16	31/64	15/32
5/8	11	.507	.201886	.0113	1 1/16	1 7/32	1 1/2	5/8	17/32	17/32
3/4	10	.62	.301907	.0125	1 1/4	1 7/13	1 49/64	3/4	5/8	5/8
7/8	9	.731	.479062	.0138	1 7/16	1 21/32	2 1/32	7/8	23/32	3/4
1	8	.837	.550226	.0156	1 5/8	1 7/8	2 19/64	1	15/16	27/32
1 1/8	7	.94	.693978	.0178	1 13/16	2 3/32	2 9/16	1 1/8	29/32	31/32
1 1/4	7	1.065	.89082	.0173	2	2 5/16	2 53/64	1 1/4	1	1 3/32
1 3/8	6	1.16	1.05683	.0208	2 3/16	2 17/32	3 3/32	1 3/8	1 3/32	1 3/16
1 1/2	6	1.284	1.29485	.0208	2 3/8	2 3/4	3 23/64	1 1/2	1 3/16	1 9/32
1 5/8	5 1/2	1.389	1.51283	.0227	2 9/16	2 31/32	3 5/8	1 5/8	1 9/32	1 13/32
1 3/4	5	1.491	1.74647	.028	2 3/4	3 3/16	3 57/64	1 3/4	1 3/8	1 1/2
1 7/8	5	1.616	2.05107	.025	2 15/16	3 13/32	4 5/32	1 7/8	1 13/32	1 5/8
2	4 1/2	1.712	2.30205	.0277	3 1/8	3 5/8	4 27/64	2	1 9/16	1 3/4
2 1/4	4 1/2	1.962	3.05424	.0277	3 1/2	4 1/16	4 61/64	2 1/4	1 3/4	1 31/32
2 1/2	4	2.176	3.71885	.0312	3 7/8	4 1/2	5 21/64	2 1/2	1 15/16	2 3/16
2 3/4	4	2.426	4.61545	.0312	4 1/4	4 29/32	6	2 3/4	2 1/8	2 7/16
3	3 1/2	2.629	5.48840	.0357	4 5/8	5 3/8	6 17/32	3	2 5/16	2 5/8
3 1/4	3 1/2	2.879	6.49890	.0357	5	5 13/16	7 1/16	3 1/4	2 1/2	2 29/32
3 1/2	3 1/4	3.1	7.54768	.0384	5 3/8	6 7/64	7 39/64	3 1/2	2 11/16	3 1/32
3 3/4	3	3.317	8.64135	.0412	5 3/4	6 21/32	8 1/8	3 3/4	2 7/8	3 11/32
4	3	3.567	9.99302	.0413	6 1/8	7 3/32	8 41/64	4	3 1/16	3 19/32
4 1/2	2 7/8	3.798	11.3292	.0435	6 1/2	7 9/16	9 3/16	4 1/4	3 1/4	3 13/16
4 1/2	2 3/4	4.028	12.8693	.0434	6 7/8	7 31/32	9 3/4	4 1/2	3 7/16	4 1/32
4 3/4	2 5/8	4.256	14.2268	.0476	7 1/4	8 13/32	10 1/4	4 3/4	3 5/8	4 9/32
5	2 1/2	4.48	15.7638	.05	7 5/8	8 27/32	10 49/64	5	3 13/16	4 1/2
5 1/2	2 1/2	4.73	17.5716	.05	8	9 9/32	11 23/64	5 1/4	4	4 3/4
5 1/2	2 3/8	4.953	19.2284	.0526	8 3/8	9 23/32	11 7/8	5 1/2	4 3/16	4 31/32
5 3/4	2 3/8	5.203	31.2535	.0526	8 3/4	10 5/32	12 3/8	5 3/4	4 3/8	5 7/32
6	2 1/4	5.423	23.0892	.0555	9 1/8	10 19/32	12 15/32	6	4 9/16	5 7/16

CHAPTER IV

CYLINDERS, CYLINDER HEADS AND FASTENINGS.

143. Cylinders with Thin Walls Subjected to Internal Pressures:—Assume a theoretically perfect cylinder, with the following notation: radius r, diameter d, length l, and thickness t. If this cylinder be subjected to an internal pressure of p pounds per square inch, it will have a tendency to cause a rupture of the shell either parallel with or transversely across its axis. In the first case the bursting force tending to split the cylinder along the line a-b is $2plr$, Fig. 53, also Church, par. 169; and the resistance to this force is the strength of the two sides, equal to $2tlf$. Forming this into an equation it gives,

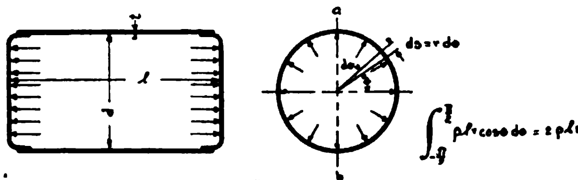


FIG. 53.

$$2tlf = 2plr, \text{ or } t = pd \div 2f. \qquad (1)$$

The force tending to cause the cylinder to fail transversely is the pressure on the head, equal to $\pi r^2 p$, and the resistance to this is the strength of the shell, equal to $2\pi rtf$ (approx.),

hence:

$$2\pi rtf = \pi r^2 p, \text{ or } t = pd \div 4f \text{ (approx.)} \qquad (2)$$

This shows that the cylinder is twice as strong to resist transverse rupture as it is to resist longitudinal rupture. Formula (2) is satisfactory for all cylinders where the shell is very thin compared to its diameter, as found in boilers. It is understood that the radius of the shell is here taken as a mean radius only. For cylinders with thick shells the resisting factor would become $\pi\ (D^2 - d^2) \div 4$ where D and d are the diameters, respectively, of the outside and the inside of the shell. Since the tendency of any cylinder, then, would be to first give way along its length, formula (1) is generally used.

144. Boiler Shell:—In applying wrought iron or steel to form cylindrical shells it is necessary to unite the ends of the plates by welding or riveting; in either case reducing the ultimate strength of the shell. Welding is resorted to in tubes and pipes, giving efficiencies varying between 50 and 90 per cent of the solid plate of the same thickness. The average efficiency may be taken at 70 per cent. Riveted joints are used on shells of large diameter such as boiler shells. The efficiencies of the joints varying, according to the Hartford Inspections and Insurance Co., between 70 per cent, for a double riveted lap joint and 86 per cent for a triple riveted butt joint having the same sized rivets. When the kind of joint is not stated, the efficiency may be taken at 70 per cent of the solid plate. For details of all forms of riveted joints for boilers see Ryersons catalog of Boiler Diagrams; Jones' Machine Design, Part II, Chap. 12; Low and Bevis, Chap. 21; Unwin, Part I, Chap. 4; The Constructor, Sec. III. Chap. 1.

The *thickness* of a boiler shell should be worked out from formula (1) by inserting the factor c as the efficiency of the joint.

Application:—What should be the thickness of a 60 inch boiler shell if the steam pressure is 150 pounds gauge and the material of the shell is open hearth steel having an ultimate tensile strength of 60,000 pounds per square inch? With a factor of safety of 6, and 70 per cent. efficiency of the joint, formula (1) gives, $t = \frac{p\,d}{2fc} = \frac{150 \times 60}{2 \times 10000 \times .70}$; $t = .643$ inches. A plate of 11-16 inches will be necessary to fulfill these conditions.

145. Machine or Cast Cylinders:—Cast iron is the material used in nearly all cases where cylinders require an inside finish for pistons or moving parts. Where the pressures are high and the thickness of the cast iron walls would be too great, steel castings are used instead of the gray iron castings. All castings are subject to hidden flaws and defects, and a great deal of uncertainty accompanies their use. Because of this uncertainty and because cylinders require reboring occasionally, designers estimate the thickness of cast cylinders by formula (1) allowing very large factors of safety, or as is the case in most *steam* and *air cylinders*, they use empirical formulas. One of the most satisfactory of these formulas for cast iron cylinders is quoted in the Trans. A. S. M. E. Vol. 18, page 741.

$$t = .05D + .3'' \qquad (3)$$

where D is the diameter of the cylinder in inches. This formula should be applied to cylinders using not over 125 pounds gauge pressure.

146. Cylinders with Thick Walls Subjected to Internal Pressures:—If the walls of a cylinder are thick compared to the cylinder diameter, the stresses in the metal, from internal pressure, will vary between the interior and exterior surfaces, being greatest on the interior fibres. The following formulas are used and give approximately the same result when designing cylinders between 6 and 18 inches diameter, and carrying pressures between 300 and 1000 pounds per square inch with a fibre stress of 2000 pounds per square inch. As the ratio of f to p becomes smaller and approaches 1 these formulas give absurd results. It is well to keep this ratio in all cases as great as 2.

$$\text{Barlow,} \qquad t = \frac{pd}{2(f-p)} \tag{4}$$

$$\text{Grashof,} \qquad t = \frac{d}{2}\left(\sqrt{\frac{3f+2p}{3f-4p}} - 1\right) \tag{5}$$

In which t = thickness of walls in inches; p=gauge pressure in pounds per square inch; d = internal diameter in inches and f = allowable fibre stress in pounds per square inch.

Another very satisfactory formula is quoted by

$$\text{Perry,} \qquad f = p\left(\frac{r_1^2 + r^2}{r_1^2 - r^2}\right)$$

Where r_1 = external radius; r = internal radius; and f and p as above.

Where a cylinder is so designed that it may be ground instead of bored it is possible to chill the inner surface to such an extent that the particles are in a state of compression and hence will yield more to the applied pressure and cause a more uniform stress throughout the shell when in service. This would make the condition of the design very similar to that of a thin cylinder. For discussions on thick cylinders see Kent, page 287; Unwin, Part 1, par. 26 a; Parry, par. 275; Cotterill, par. 214.

147. Cylinders with Thin Walls Subjected to External Pressures:—An empirical formula developed by Fairbairn for the collapsing pressure of thin wrought iron tubes is discussed in Cotterill, par. 181; Unwin, Part. 1, par. 41, and Kent, page 264. Let p=differential pressure in pounds per square inch, t = thickness of shell in inches, d = diameter in inches and l = length in inches, then

$$p = 9672000\ t^2 \div ld \tag{6}$$

This should not be used for extreme cases nor for thicknesses less than 3/8 of an inch.

Lloyds Register contains the following formula,

$$p = 1075200\ t^2 \div ld \tag{7}$$

It will be seen that this is the same as (6) with a factor of safety of 9.

148. A Thin Sphere Subjected to Internal Pressure would have a tendency to rupture along the circumference of its diameter. The force exerted would be $p\pi d^2 \div 4$ and the resistance $\pi d\,t\,f$ approximately, hence we have with the same notation as above,

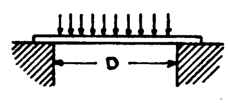
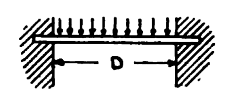
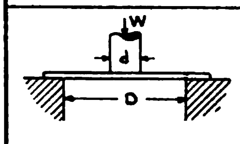
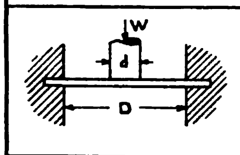
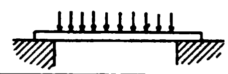
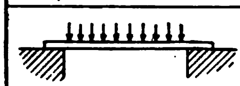
$$t = pd \div 4f \tag{8}$$

It will be seen that (8) is the same as (2) and makes the sphere twice as strong as the cylinder of the same diameter and thickness of shell; hence a spherical boiler head would be as strong as the shell if it was one-half its thickness. When this formula is applied to wrought iron or steel work it will be necessary to account for the efficiency of the riveted joint.

Dished heads, for boiler work, have the same strength to resist rupture as the boiler shell of the same thickness, if the radius of curvature is equal to the diameter of the boiler shell. This may be obtained by inspection from formulas (1) and (8). The *camber* of the head will be $h = .134\ d$; where d = diameter of shell and h = height of camber, both given in same units. Low and Bevis, page 305.

149. Strength of Flat Plates:—The relation existing between the thickness of a flat plate and its fibre stress and deflection when subjected to known conditions of loading can be obtained from the formulas given in Table XVIII. See also, American Machinist, 1906, page 683.

TABLE XVIII.

	CONDITION	AUTHOR	FIBRE STRESS f	DEFLECTION Δ	NOTES.
	Flat circular plate. Uniform load. Supported at edge.	GRASHOF	$f = \frac{5}{24} \frac{D^2 w}{t^2}$	$\Delta = \frac{1}{24} \frac{D^4 w}{t^3 E}$	t = Thickness of plate w = Load in # per □"
		MERRIMAN	C. I: $f = \frac{3}{8} \frac{D^2 w}{t^2}$ W. I and S.: $f = \frac{1}{5} \frac{D^2 w}{t^2}$		
	Flat circular plate. Uniform load. Fixed at edge.	GRASHOF	$f = \frac{1}{6} \frac{D^2 w}{t^2}$	$\Delta = \frac{1}{96} \frac{D^4 w}{t^3 E}$	
		MERRIMAN	C. I: $f = \frac{9}{32} \frac{D^2 w}{t^2}$ W. I and S: $f = \frac{1}{4} \frac{D^2 w}{t^2}$		
	Flat circular plate. Concentrated load within circumference πd. Supported at edge	GRASHOF	$f = \left[\frac{4}{3} \log_e\left(\frac{D}{d}\right) + 1\right] \frac{W}{\pi t^2}$ $f = C \frac{W}{\pi t^2}$	$\Delta = .13 \frac{D^2 W}{t^3 E}$	W = Total load D/d = 10, 20, 30, 40, 50 C = 4.07, 5.0, 5.63, 5.92, 6.22
		MERRIMAN	$f = \frac{2w(1-a)}{t^2}\left(\frac{Dd}{2} - \frac{d^2}{4}\right)$		w = Uniform pressure in # per □" over area "d". For C. I $a = \frac{1}{4}$ For W I and Steel $a = \frac{1}{3}$.
	Flat circular plate Concentrated load Within circumference πd. Fixed at edge.	GRASHOF	$f = \left[\frac{4}{3} \log_e\left(\frac{D}{d}\right)\right] \frac{W}{\pi t^2}$ $f = C \frac{W}{\pi t^2}$	$\Delta = .052 \frac{D^2 W}{t^3 E}$	W = Total load D/d = 10, 20, 30, 40, 50 C = 3.07, 4.0, 4.53, 4.92, 5.22
		MERRIMAN	$f = \frac{3w(1-a)}{2t^2}\left(\frac{Dd}{2} - \frac{d^2}{4}\right)$		w = Uniform pressure in # per □" over area "d". For C.I–a = $\frac{1}{4}$. For W I & Steel–a = $\frac{1}{3}$.
	Elliptical flat plate. Uniform load. Supported at edge. Semi-axes A and B.	BACH MERRIMAN	$f = a\frac{A^2 B^2 w}{(A^2 + B^2) t^2}$		For C.I–a = 3. For W I & Steel–a = $\frac{9}{4}$. For fixed plates:—a for C I = $\frac{9}{4}$; For W.I & S–a = 2.
	Rectangular flat plate. Uniform load. Supported or fixed. Sides L and M.	BACH MERRIMAN	$f = a\frac{L^2 M^2 w}{(L^2 + M^2) t^2}$		$a = \frac{9}{4}$ for edges supported. $a = \frac{3}{2}$ for edges fixed

Of all the conditions here shown the second one is the one most often found in practice. This formula would cover most forms of cylinder heads. Kent discusses the subject on pages 284 and 794 of his Pocket Book and suggests a number of empirical rules to be used instead of the rational ones, which he says, in many cases give too large results.

Where approximate values are considered satisfactory, it is good practice to take the thickness of the head at the edge 25 per cent greater than the thickness of the cast iron cylinder walls. The central part of the head may be made thinner than the edge, and is sometimes set off from the plane of the outer flange. The thickness of this central portion should be about the thickness of the cylinder walls.

Where *extra heavy* pressures are involved, it is not safe to depend upon the approximate rules, but each case should be worked out by an approved formula, and then checked up by the best current practice.

150. Method of Fastening Between Cylinder and Cylinder Head:—In most cylinder work it is necessary to cap the ends. The form of this capping differs according to the materials employed and the use to which the cylinder is to be applied. The wrought iron or steel cylinder, for example, is used mostly on boilers and is capped with a head of like material. This head is flanged and riveted to the shell. If the head is dished, as shown in Fig. 54 A, to the radius of the shell diameter, the thickness of the material may be the same as that of the shell and need not be stayed, but if the head is flat as in B, it is made somewhat thicker than the shell and in addition is stayed. The staying may be done by *through stays* or by *angle stays* depending upon the interior construction and requirements of the boiler. For more details on staying see notes on Engine and Boiler design.

Cast Cylinders for steam engines, compressors, hoists, etc., would have a fastening similar to C, the heads being bolted to flanges that are an integral part with the cylinder body. These fastenings may be by ordinary bolts, stud bolts, or cap screws. For rough work the bolt is preferred. For finished work the stud bolt is preferred. The cap screw would be used only in special cases where the cylinder head would seldom be removed, since frequent removals would tend to ruin the threads in the cast iron flange.

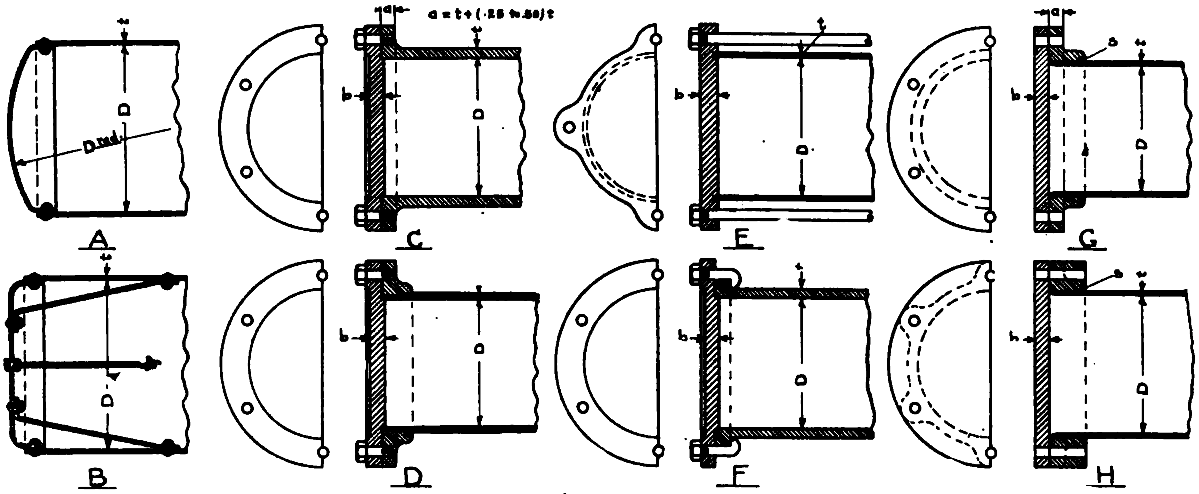


FIG. 54.

Concerning the *number* and the *size of the bolts*; it is better to have a number of bolts of small cross sectional area, than a less number of bolts of large area, for if one bolt proves defective the extra stress is more readily absorbed by those near to it. The number of bolts used will approximate closely to the nominal diameter of the cylinder in inches, i. e. a 6 inch cylinder, 6 bolts; a 24 inch cylinder, 24 bolts, etc.

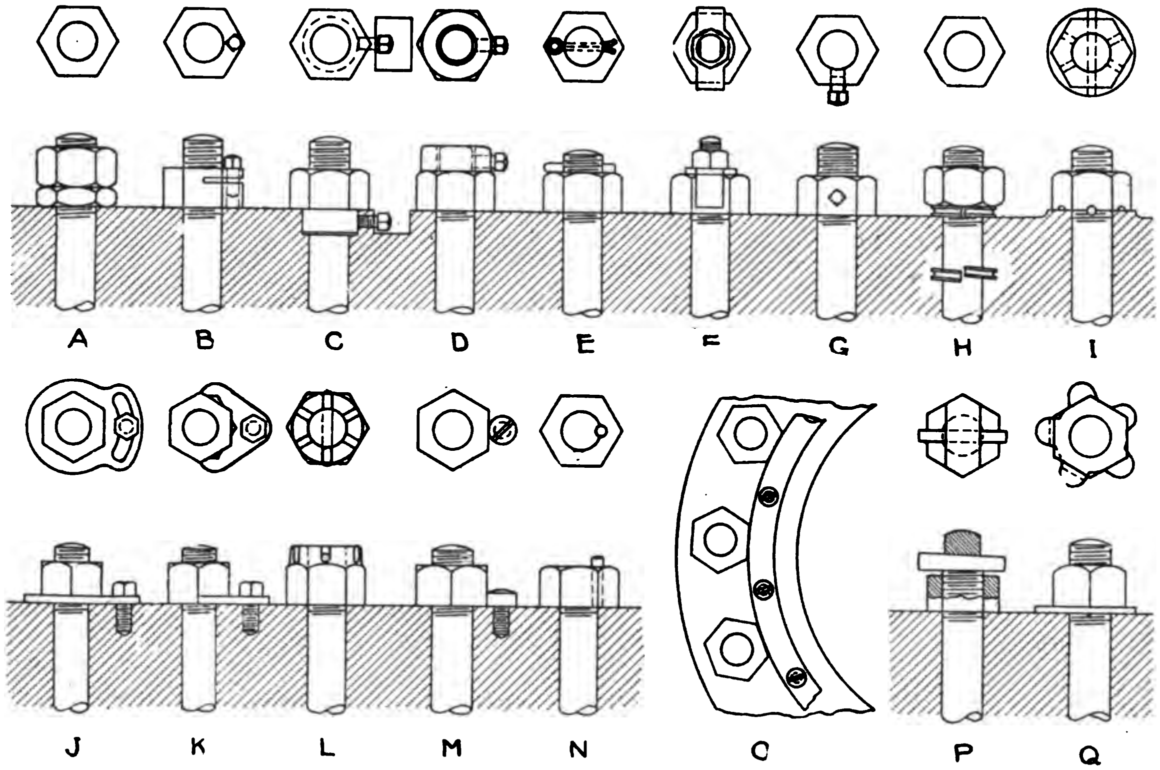
In cylinders carrying a pressure of 75 to 100 pounds gauge, it is not advisable to use less than a ⅝ inch bolt because of the liability of injury when setting the nut. All bolts should be figured at the root of the thread. All bearing surfaces should be acurately finished. The joint should be tight under pressure, either by being ground to a fit or by facing with some soft thin packing.

A modification of C is shown in F where eccentric head bolts are used instead of the ordinary stud bolt.

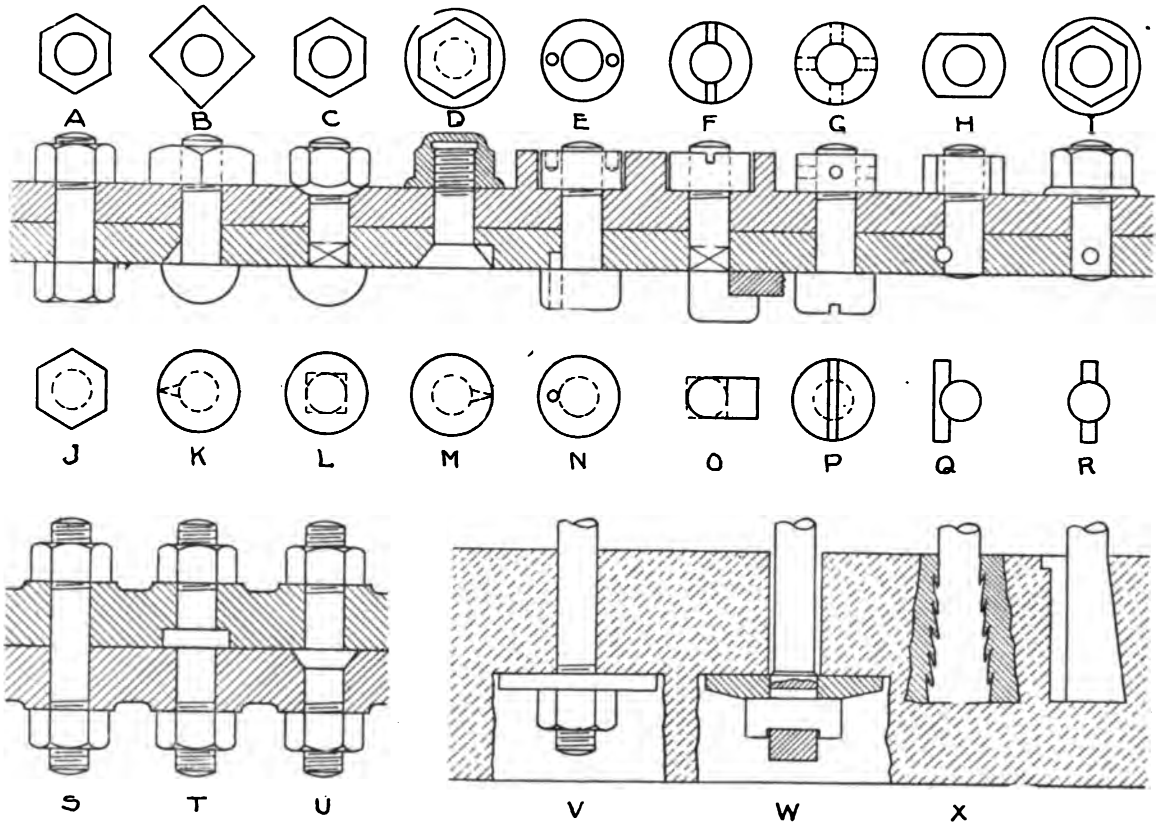
To cap a large pipe cylinder with a cast iron head it is necessary to first fasten a flange to the pipe by screwing or expanding the pipe into it, and then fasten the head to this flange as shown in *D*, *G* and *H*. In *D* the pipe should be screwed through the flange and faced off with it. *G* and *H* should be faced off in like manner and in addition should have soft metal packing swaged into dovetail grooves as at *s*.

A *Brass* or *Copper tube cylinder* may be capped as shown at *E*. This metal being too thin to be threaded it is faced off square at each end and bolted between cast heads. A soft copper or leád packing may be applied to the bottom of each finished groove to cut off leakage. Such cylinders are very common in small air hoists.

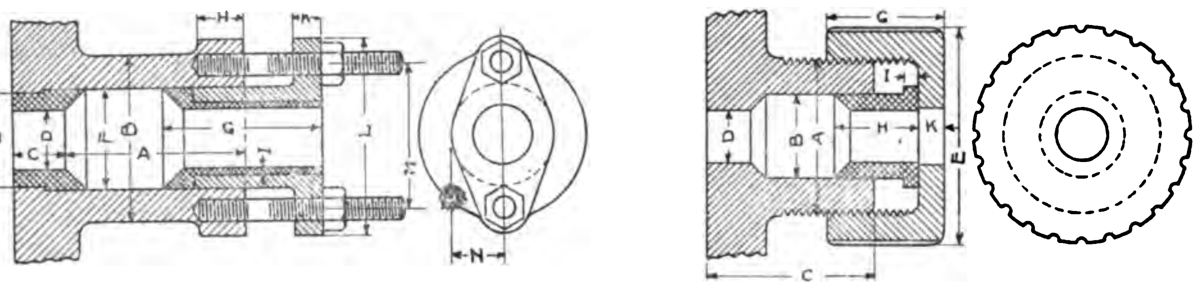
151. Nut Fastenings, Fig. 55:—



152. Bolt Fastenings, Fig. 56:—



153. Standard Sizes of Stuffing Boxes, Fig. 57:—



D = DIAM. OF ROD
A = 1.6D + 1.5″
B = 1.75D + 1.125″
C = .1D + .75″
E = 1.25D + .375″
F = 1.25D + .625″
G = 1.5D + 1.″
H = .3D + .5″
I = .04D + .1875″
K = .25D + .25″
L = 2.25D + 1.75″
M = 1.6D + 1.25″
N = .75D + .375″

USE 2 BOLTS IN GLAND ON RODS UP TO 3.5″ DIAM. ABOVE THAT SIZE MAKE GLAND ROUND AND PUT IN 3 BOLTS.

D = DIAM. OF ROD
A = 2.5D + .5″
B = 1.5D + .125″
C = 3D + .25″
E = 3.5D + .625″
G = 2D + .25″
H = 1.5D + .25″
I = .25D + .0625″
K = .5D

THIS STYLE USED ON RODS UP TO 1¼″ DIAM. THREADS PER INCH SAME AS FOR BOLT OF DIAM. OF ROD.

CHAPTER V.

DESIGN NO. 1.

NOTES RELATING TO THE DRAWINGS.

154. Size of Drawings:—The following standard dimensions are given as the cutting sizes of the sheets. The designer is at liberty to make his own selection from these sizes. It is suggested however, that the sheets be taken as small as will admit of a clear and distinct set of drawings.

24″ × 36″—Size A
18″ × 24″— " B
12″ × 18″— " C
9″ × 12″— " D

Scale:—Any scale may be taken which will show clearly all the details and give a good arrangement on the sheet. Details may have different scales on the same sheet if so desired. When this is done each detail should have the scale given.

Border Line:—A margin of one quarter of an inch should be left between the border line and the edge of the finished sheet.

Name Plate:—Make the name plate or title at the lower right hand corner to cover a space 2¼′ × 3½″ inside the border line. This name plate should be considered standard and should be worked up after the plan of those shown on the following plates. It would be well for each designer to make a standard corner plate to be used below the various tracings when working up this part.

All drawings will be worked up completely in pencil and turned in to the instructor. The instructor will give them to another designer who will be responsible for the checking. This checking will be done in the form of notes on a separate paper and attached to the drawing. These are then returned to the designer for approval and corrections. The designer then traces his drawings and after obtaining the signature of the checker to them submits the same with the checkers' notes to the instructor for approval.

In all this work Jamison's Mechanical Drawing will be used as reference concerning arrangement of views, sectioning, cross hatching, lettering, and the like.

Every dimension should be clearly shown so that no measurements need be taken by scale from the drawing.

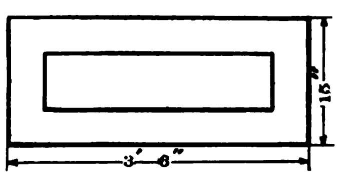
All dimensions should be given in round vertical figures, heavy enough to print well. No diagonal-barred fractions, thin or doubtful figures will be accepted.

All dimensions below 3′ — 0″ should be given in inches.

All dimension lines should be made as light as will insure good printing and should have a central space for figures.

All dimensions should read in the direction of the arrows.

Avoid crowding the dimensions to the center of any detail. A much better way is by the use of projected lines as shown.



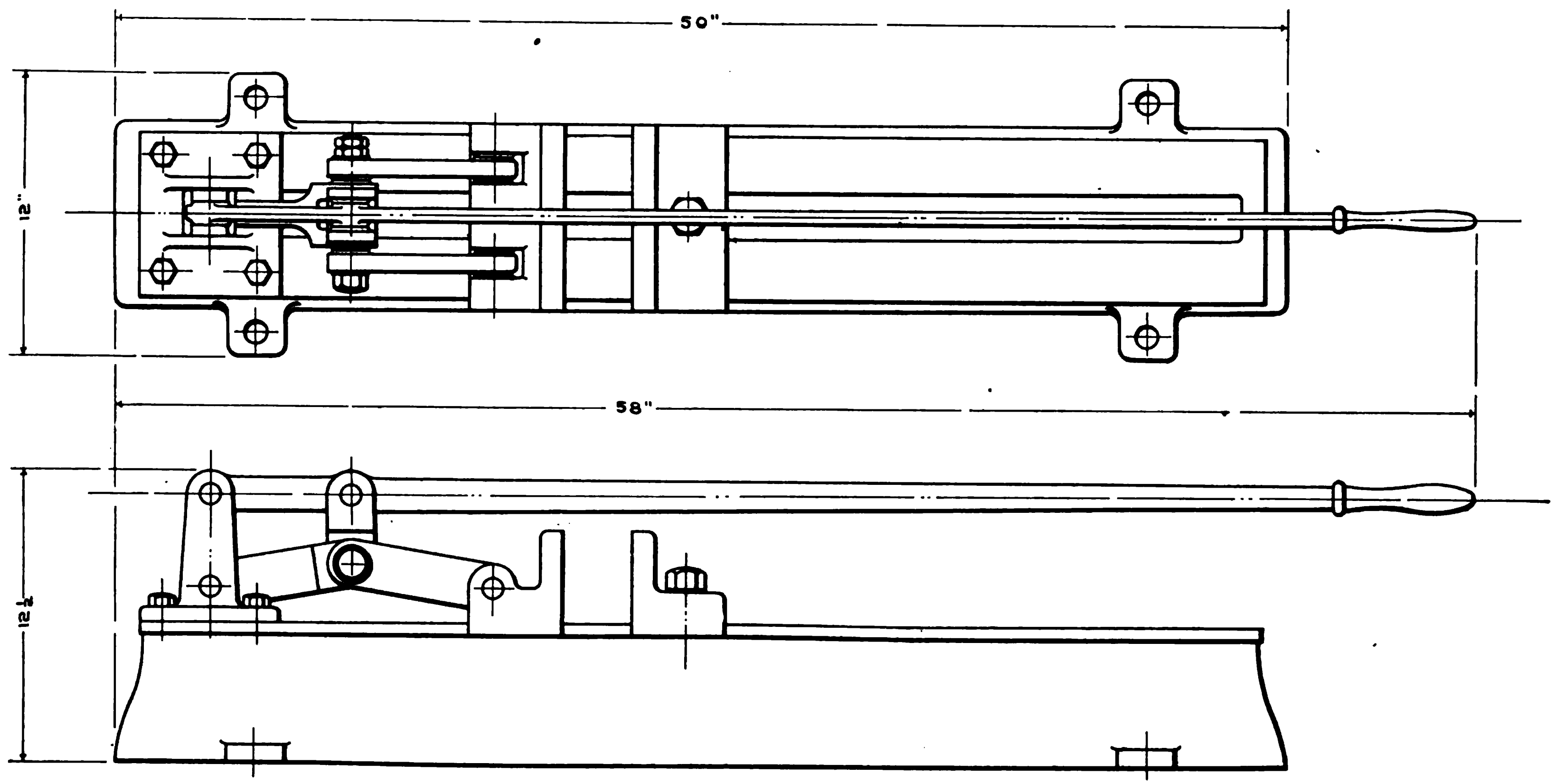
All detailed pieces should be accompanied by a *shop note* or *call* as "C. I. One wanted;" "M. S. Two wanted"; "Finish all over," "Turned for a shrinking fit", etc., etc.

The following abbreviations will be considered satisfactory in these calls:—

C. I.........Cast Iron.
C. S.........Cast Steel.
W. I........Wrought Iron.
M. S........Machine Steel.
B. b. t......Babbitt Metal.
F. ....Finish.
D. ....Diameter.
R. ....Radius.

155. Calculations:—Each designer is expected to draw up a report in parallel with the design. This report will contain free-hand sketches and a brief summary of calculations and accepted sizes of the different parts of the design, and will be submitted with the finished tracings.

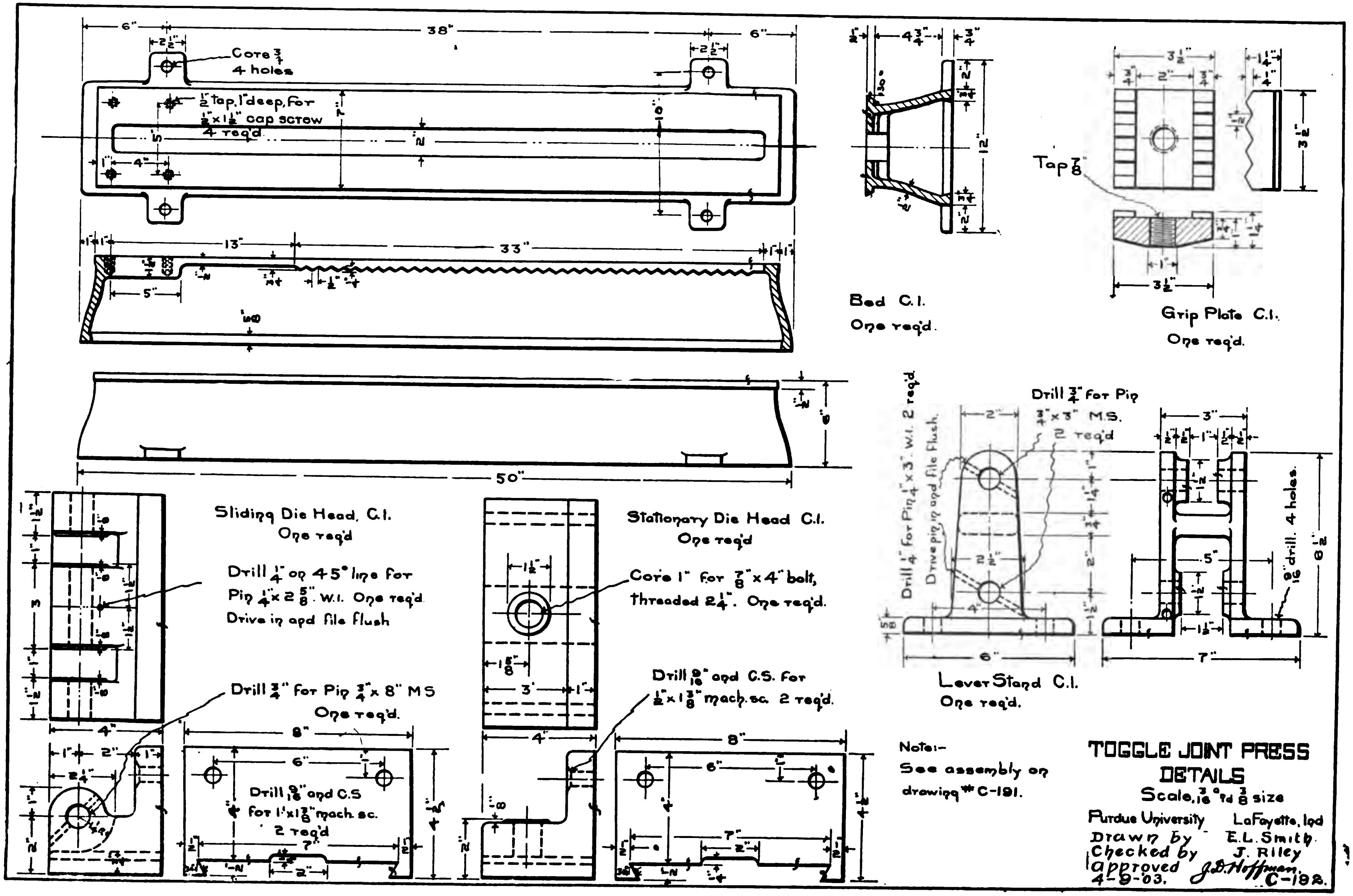
156. Checking:—Each designer should have experience not only in planning and executing well his own designs, but he should take up designs of other men and offer suggestions and criticisms upon their work. The way to obtain this experience has been suggested above.



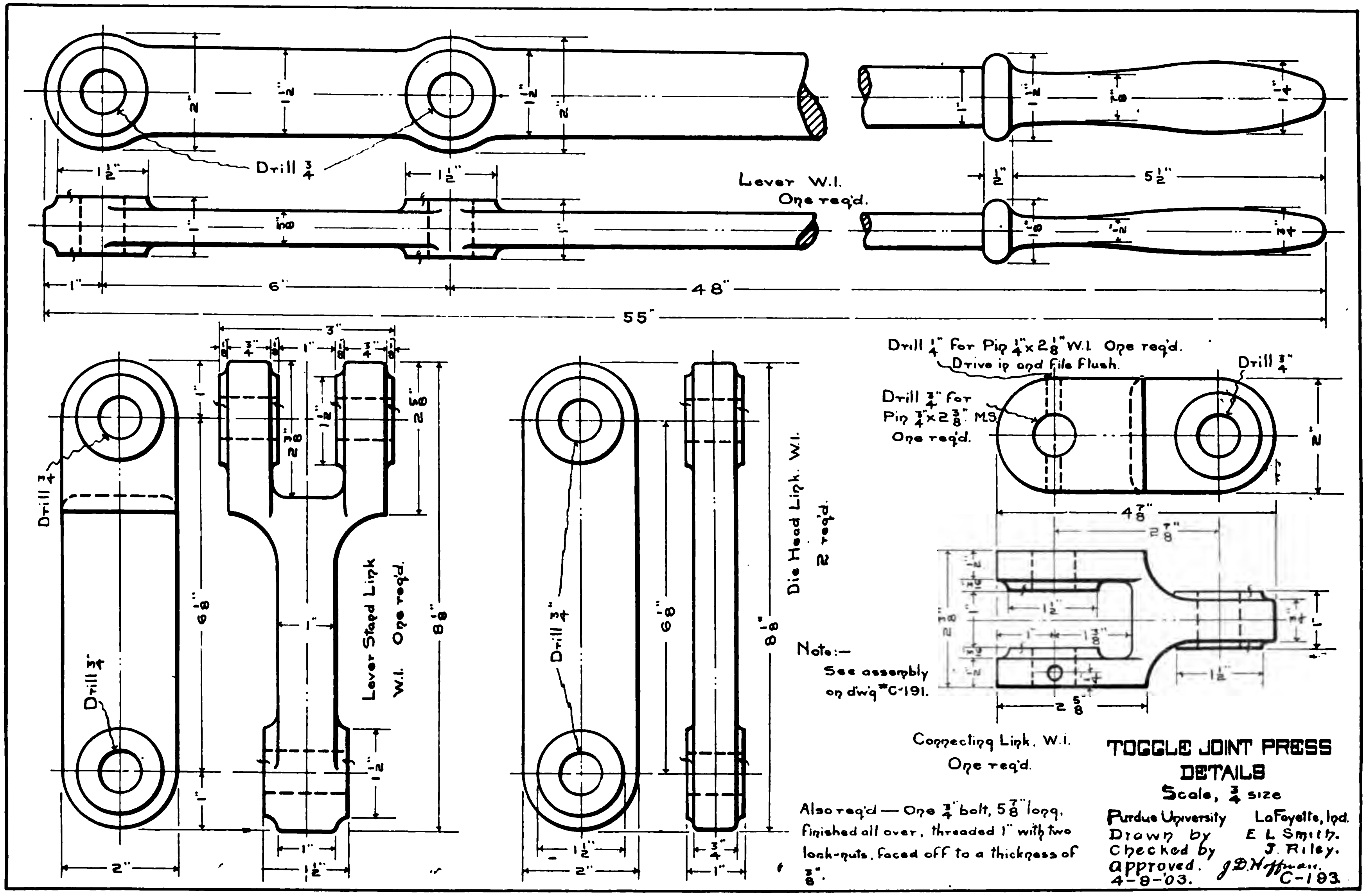
Note: See details on drawings C-192 and C-193.

TOGGLE JOINT PRESS
ASSEMBLY
Scale – ¼ size.

Purdue University LaFayette, Ind.
Drawn by E.L. Smith.
Checked by J. Riley.
Approved J.D. Hoffman.
4-8-03. C-191.



Core 3/4 4 holes
1/2" tap, 1" deep, for 1/2" x 1 1/2" cap screw 4 req'd.
Bed C.I.
One req'd.
Tap 7/8"
Grip Plate C.I.
One req'd.
Drill 1/4" for Pin 1/4" x 3" W.I. 2 req'd.
Drive pin in and file flush.
Drill 3/4" for Pin 3/4" x 3" M.S. 2 req'd
9/16" drill. 4 holes.
Lever Stand C.I.
One req'd.
Sliding Die Head, C.I.
One req'd
Drill 1/4" on 45° line for Pin 1/4" x 2 5/8" W.I. One req'd
Drive in and file flush
Drill 3/4" for Pin 3/4" x 8" M.S
One req'd.
Drill 9/16" and C.S for 1" x 1 3/8" mach. sc. 2 req'd
Stationary Die Head C.I.
One req'd
Core 1" for 7/8" x 4" bolt, threaded 2 1/4". One req'd.
Drill 9/16" and C.S. for 1/2" x 1 3/8" mach. sc. 2 req'd.
Note:- See assembly on drawing # C-191.
TOGGLE JOINT PRESS
DETAILS
Scale, 3/16" and 3/8 size
Purdue University LaFayette, Ind
Drawn by E.L. Smith
Checked by J. Riley
Approved J.D. Hoffman
4-9-03.
C-192.


Lever W.I.
One req'd.
Drill 3/4
55"
48"
6"
5 1/2"
Lever Stand Link
W.I. One req'd.
Drill 3/4"
8 1/8"
6 1/8"
Die Head Link. W.I.
2 req'd.
Drill 1/4" for Pin 1/4" x 2 1/8" W.I. One req'd.
Drive in and file flush.
Drill 3/4" for
Pin 3/4" x 2 3/8" M.S.
One req'd.
Note:—
See assembly
on dwg #C-191.
Connecting Link. W.I.
One req'd.
Also req'd — One 3/4" bolt, 5 7/8" long, finished all over, threaded 1" with two lock-nuts, faced off to a thickness of 3/8".
TOGGLE JOINT PRESS
DETAILS
Scale, 3/4 size
Purdue University LaFayette, Ind.
Drawn by E L Smith.
Checked by J. Riley.
Approved. J.D. Hoffman.
4-9-03.
C-193

In checking up the work of another man the following points should be observed:

(1). General appearance of the design relative to workmanship and execution, arrangement of drawings, notes, dimensions, etc.

(2). General design relative to proportion, and strength and arrangement of parts. This is to be merely the checker's impression and need not require checking of original calculations.

No drawing should be retained longer than one exercise and at the completion of the checking should be returned to the designer. It is estimated that any set of drawings may be checked in this way within two hours time. No notes or marks will be made on the drawings but special paper will be provided for this purpose. In looking over the drawings finally, the instructor will give credit to the work of the checker as well as to that of the designer.

ANALYSIS OF THE DESIGN.

We are now prepared to take up the first design, the Toggle Joint Press. Machines of this class are sometimes used in forming thin sheets of copper and brass into articles for ornamental purposes, consequently it is a useful tool. Plates C-191, C-192, and C-193 show a design of one of the smaller machines and are inserted to give an idea as to the scope of the work. The design was worked up on three 12″ × 18″ sheets; two of details and one assembly view.

It is urged that the designer regard these sheets merely as illustrative of a good drawing room job and that, from the standpoint of ideas of design he will cultivate originality and make his design as nearly independent as possible.

157. Forces Involved:—Referring to Plate C-191, it will be seen that the forces involved can be represented in the following force diagram, with the direction of the forces represented by arrows.

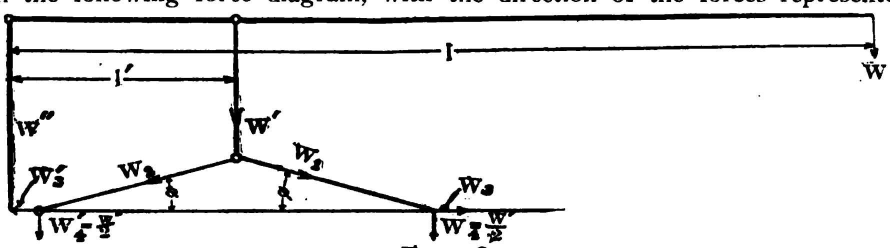


FIG. 58.

Each designer will be given a value for W, l, l' and Φ. In all the designs Φ will be taken at 10°, assuming that the maximum load will be carried at this position and that the lever arm will then be horizontal.

In the assignments the range of values will be as follows:—

$W = 200, 300, 400 \ldots\ldots 1000$

$l = 4', 4'\,6'', 5', 5'\,6'' \ldots\ldots 10'$

$l' = \begin{cases} 6'', 8'', 10'' \text{ and } 12'' \text{ for the larger sizes.} \\ 6'' \text{ to } 8'' \text{ for the smaller sizes,} \end{cases}$

To make the analysis plain one assignment will be worked through and brief comments made upon it. Selecting $W = 100\#$; $l = 5' - 3'' = 63''$; $l' = 7''$ and $\Phi = 10°$ we have the following from the force diagram:

$$W' = \frac{W\,l}{l'} = \frac{100 \times 63}{7} = 900\#$$

$$W'' = \frac{W\,(l - l')}{l'} = \frac{100 \times 56}{7} = 800\#$$

$$W_1 = W_2 = \frac{W'}{2 \sin.\,\Phi} = \frac{900}{.3473} = 2591.4\#$$

$$W_3 = W_1 \cos.\,\Phi = 2591.4 \times .98481 = 2552\#$$

$$W_4 = \frac{900}{2} = 450\#$$

158. Lever:—This will be designed as a beam in flexure, par. 46. The designer must here decide if the beam is to have parallel sides in which case b in the modulus for the rectangular section would be constant, or taper sides in which case a certain ratio of $b \div h$ would be used. The best way to decide which to use is to find the size of the sections at g and c for each case. Assuming $f = 8{,}000$ for wrought iron, $b = 1$ and disregarding the hole at c which has little effect, our formula $W = f\,Z$ becomes

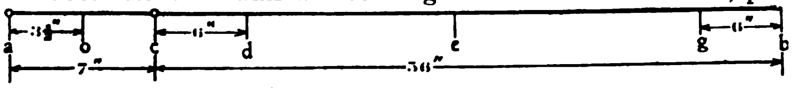


FIG. 59.

$100 \times 6 = 8{,}000\, h^2 \div 6$; $h = .67''$ at g and

$100 \times 56 = 8{,}000\, h^2 \div 6$; $h = 2.05''$ at C.

This beam would have a better shape and would also be lighter if the thickness be reduced below 1″, say to ¾″. With this value the formula becomes

$100 \times 6 = 8{,}000\, h^2 \div 8$; $h = .77''$ at g and

$100 \times 56 = 8{,}000\, h^2 \div 8$; $h = 2.37''$ at C.

The values thus obtained would give a very good shaped beam having a section $.75'' \times .77''$ at g and $.75'' \times 2.37''$ at C.

On the other hand, suppose a ratio of $b \div h = \frac{1}{4}$ to be desired, the problem becomes

$100 \times 6 = 8{,}000\, h^3 \div 24$; $h = 1.22''$ and $b = 1.22 \div 4 = .3''$ at g.

$100 \times 56 = 8{,}000\, h^3 \div 24$; $h = 2.56''$ and $b = 2.56 \div 4 = .64''$ at c.

section at $g = .3'' \times 1.22''$

section at $C = .64'' \times 2.56''$

The above gives the methods of determining the size of the section at any point of the beam. Sections should be taken at regular intervals of length and a diagram plotted from the results. One section only need be taken between a and c, say at o midway between. This diagram when completed will show the beam to take the form of a curve similar to Fig. 60. It will be found convenient however, to approximate this curve with a straight line as $x\ y$. This would be satisfactory for strength and would be more easily constructed.

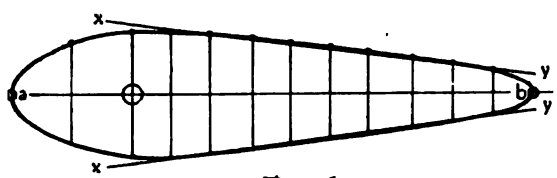


FIG. 60.

It will be noticed that the bending moment becomes zero at the points a and o where the loads are applied. This would theoretically give no size to the handles and make it impossible of construction. Some satisfactory design of handle or hub must be arranged for at these points of sufficient size to carry the pins or bolts, each having the sides and edges of the beam filleted into them in a neat manner. See Plate C-193. A handle can be placed at b for all loads of 300 pounds or less and a drilled hub for larger loads so that a small air or steam cylinder can be attached. A similar hub will be added at a, for connection to the post at the rear.

159. The following shapes may be found useful in designing the lever.

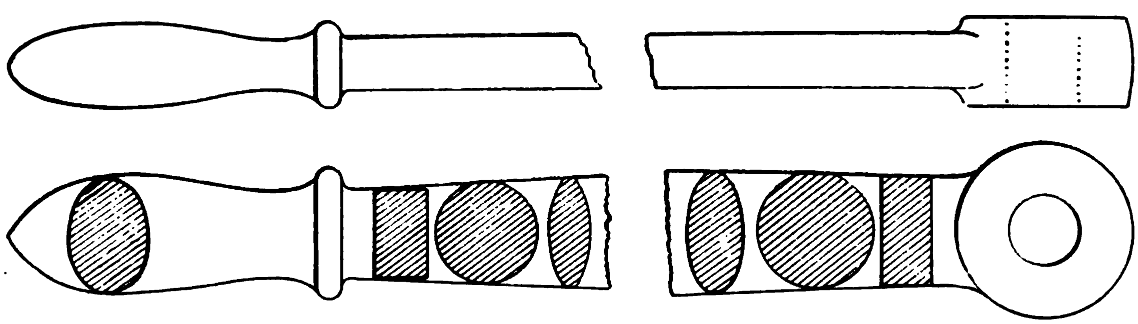


FIG. 61. Shapes at b.

The size and shape of the handle or hub at this end will be largely a matter of looks since the load carried is very small. The pin if one is used may be calculated for double shear to get the minimum size allowable, but this size will probably be so small that it will be necessary to increase the size of both pin and hub to add neatness in design. Such points as this call for special investigation. Any piece of a machine may be made extra strong if desirable, to harmonize with the other parts of the machine, but the reverse is not the case.

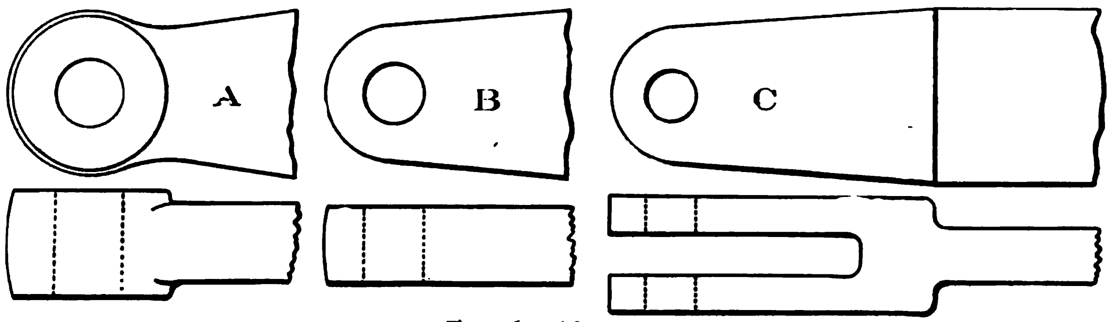


FIG. 62. Shapes at *a*.

In the above shapes for this end *A* and *B* would be preferred because of the ease of construction. In most cases the post which connects with this would be made of cast iron and could easily be cored out to fit over the lever arm end rather than to fit the arm end over the post. The only calculations necessary here are those that determine the diameter of the hub and the length of the hub, of the lever. Make the diameter of this hub equal to the diameter of the cast hub of the standard. To illustrate: At *a*, a tensional force of W'' is acting and this force is resisted by four areas on the section, *R S*, equal in total area to R' S', of the standard. These sections are figured for cast iron in direct tension by the formula $W = f\ A$. The four areas on *R S*. are alike and the ratio of $b \div h$ may be assumed. Having figured the pin for double shear by the formula $W'' = 2\ f\ A$ find the diameter of the pin and add to it $2\ h$, this will give the diameter of the cast hub and consequently the diameter of the lever end.

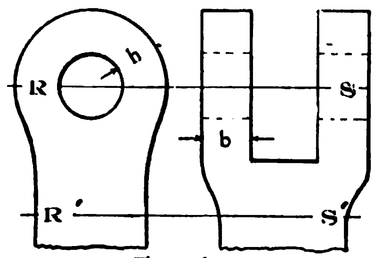


FIG. 63.

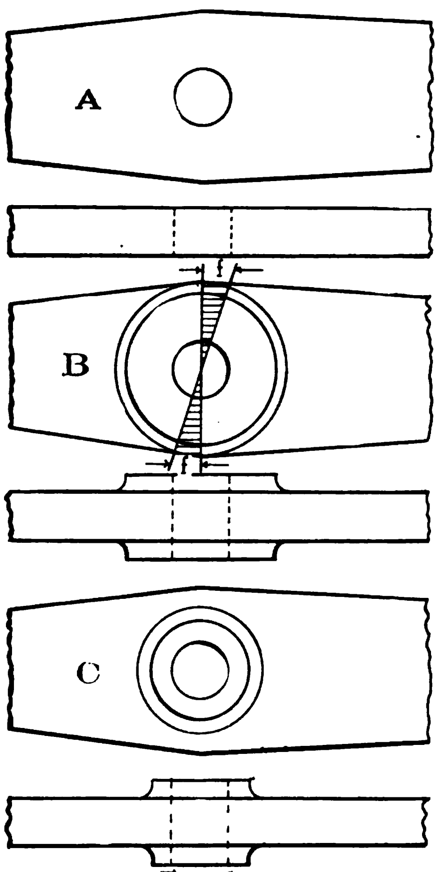


FIG. 64.

160. If *f* for shear in wrought iron be taken at 5000#□″, the diameter of the pin will be .33 inch or say $\frac{3}{8}$ inch. If *f* for tension in cast iron be taken at 1500#□″, the area *b h* will be .133 square inch, from which, if *b* be taken at $\frac{1}{4}$ inch, *h* becomes .53 inch. This would make the diameter of the hubs at *a* $1\frac{3}{8}$ inches.

It would be next in order to find if the wrought iron pin will crush under the load applied. The part of the pin in the casting and the part in the lever is each subjected to a crushing action. The resistance of the pin to crushing is in proportion to the projected area of the pin, and for the part in the casting is $2\ b\ d = 2 \times \frac{1}{4} \times \frac{3}{8} = 3\text{-}16$ square inch. Which, if $f_c = f_s = 5000$, will sustain $\frac{3}{16} \times 5000 = 938$ pounds. If it had been found that *b h* was so small that the load it was capable of supporting was less than the load applied, then *b* would have to be increased, and if desired, *h* could be diminished to retain the same total area. The thickness of the lever arm end should be at least as much as $2\ b = \frac{1}{2}$ inch in this case. . From inspection, however, it is readily seen that the thickness at *a* would be necessarily increased to correspond to the thickness at $c = .64$ inch.

In every fastening of this kind, it would be well to investigate the shearing of the pin, the tension of the hub members and the crushing of the pin.

In calculating the size of the section at *c* Fig. 59, the hole was not considered. The error introduced by this is very slight and in most cases may be neglected. The fibre stress in the cross section

of the arm varies from zero at the center to a maximum at the edge as shown in the diagram B, Fig. 64, where by proportion we can readily obtain the relative resistance offered by the metal at the center as compared to that at the edge of the section. The loss at the center is more than taken up by the addition of a fraction of an inch at the edge or a very small boss around the hole. If the hole in any case should be large, a modulus could be selected for this hollow section, and the exact figures obtained.

The pin would be calculated in double shear.

The size of the boss, if any be added, is largely optional.

161. Standard:—The design of the standard would depend largely on the magnitude of the force to be resisted. In the smaller sized machines it would undoubtedly be made of cast iron and as such the upper end would be as shown in the preceding paragraph. In the larger sized machines the standard would be made of wrought iron or steel plates in which case the size of the hub would be figured with a different value of f.

The cross section of the body of the standard may be shaped as in Fig. 65. Assuming the areas to be equal, the strongest section to resist any bending action that may come up it, is D.

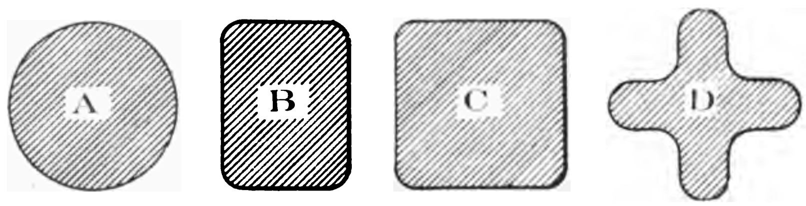


FIG. 65.

The lower end of the standard would be cored out to receive the rod W_2, and would have a flange for fastening to the top of the bed. Fig. 66 shows some of the shapes that may be used.

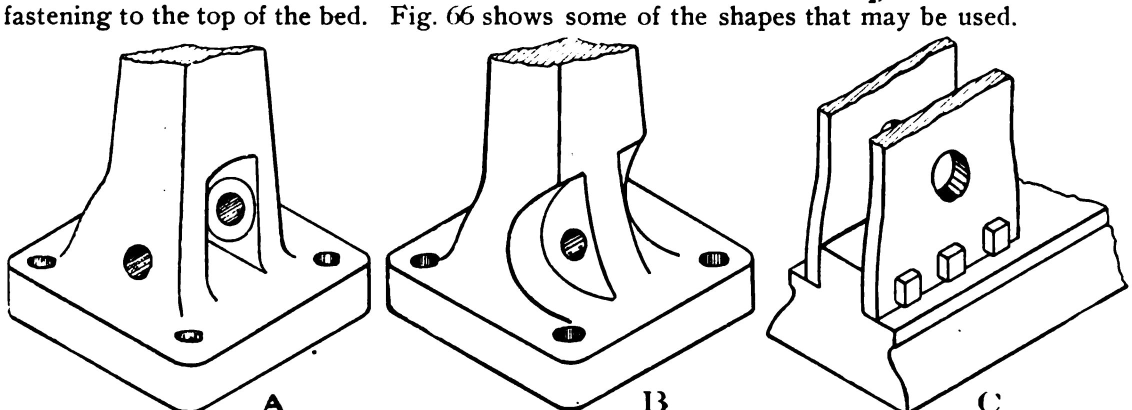


FIG. 66.

The pin at the base is figured for double shear by the formula $W_2 = 2 f A$.

162. In deciding upon the kind of fastening between the standard and the bed, it would be well to first examine it regarding the turning moments about a, Fig. 67.

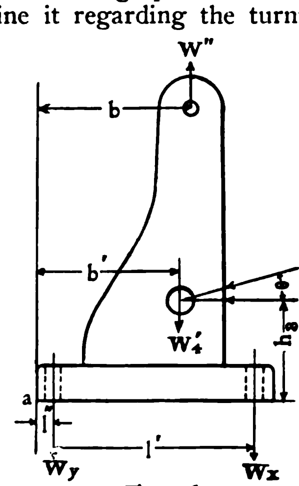


FIG. 67.

$$W'' b + W_3 h_3 - W'_4 b' = W_x l' + W_y l''$$

Assume $b = b' = 3''$, $h_3 = 2''$, $l' = 5''$ and $l'' = 1''$ then with $W'' = 800$, $W_3 = 2552$, $W_4' = 450$. We have $5\ W_x + W_y = 6154$.

If $W_x = W_y$ then $6\ W_x = 6154$ or $W_x = 1026\#$. This is equivalent to a ½ inch bolt. Suppose W_y, because of its location, to be of little value in resisting turning about a, then $5\ W_x = 6154$. $W_x = 1231\# =$ approx. $\frac{9}{16}''$ bolt. If more than one bolt is used the total bolt area may be the equivalent of that given above.

Next examine the joint for a summation of all vertical forces.

$W'' - W_4' = W_x + W_y$. If $W_x = W_y$ then, $2\ W_x = 800 - 450 = 350\#$ $\therefore$ $W_x = 175\#$.

Since this force is less than that obtained by moments it need not be considered.

Next examine the joint for a summation of the horizontal forces. In this the force W_3 tends to shear the bolts or to shear the casting inside the bolt holes. In the first

$$W_3 = f A. \text{ If we take } f = 5000, \text{ then,}$$

$$2552 = 5000\, A. \quad \therefore \quad A = .51^{\square''} \text{ of bolt area.}$$

If the bolt shears at the root of the thread as would be the case with a cap screw we have at least four ½ inch cap screws needed.

In the second case if the flange is say 6 inches long we have

$$2552 = 2 \times 6\, t\, f. \text{ Let } f = 1500 \text{ for cast iron we have } t = \text{approx. } .15''.$$

This would, of course, be made thicker say ½ inch for the appearance and looks of the casting.

In the above discussion of the standard fastening the tendency to failure would apparently be the shearing of the bolts. This would not be true in every case; for example if h_3 were greater the failure of the joint would probably be by moments about a. The calculations would be modified, also by the arrangement of the bolts or cap screws.

It is well in every case to examine a joint from all standpoints and use the greatest requirement.

163. Toggle:—In designing the toggle there are two ways in which it may fail at the joint: by shearing the pin, and by bending the pin. In Fig. 68, (A) and (B) show very simple arrangements of this joint. To obtain the size of the pin from shear in this case

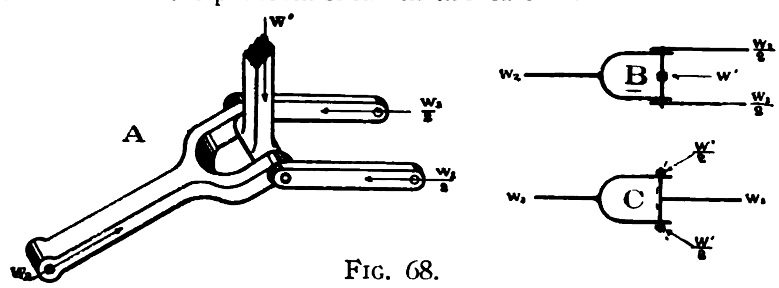


FIG. 68.

$$W_1 = W_2 = 2 f A. \qquad \text{If } f = 5000, \text{ then}$$

$$2591.4 = 2 \times 5000\, A.; \qquad A = .26^{\square''};\ d = .58'' \text{ say } \tfrac{5}{8}''.$$

It is readily seen that the pin would be found to be the same size if the load $W_1 \div 2$ were figured for single shear.

To obtain the size of the pin to resist bending assume some length of pin between the outer forces $W_1 \div 2$ as 2″ and solve by the formula

$W' l \div 8 = f Z$. If $f = 8000$ then $900 \times 2 \div 8 = 8{,}000 \times \pi d^3 \div 32 \ \therefore\ d = .65$ say $\frac{11}{16}''$.

With the toggle acting on one pin at the center as shown, the smaller force W' should come at the center of the length of the pin as shown in A and B. If the heavier force W_1 or W_2 act at the center of the pin it would have an unnecessary bending strain as shown in C, and would require too large a pin.

164. Fig. 69 gives some shapes of toggle members. A, B, C and D are usual shapes of the horizontal members. A and B have split ends and are necessarily hard to forge and machine. C is the simplest form. This form is sometimes modified by adding bosses to one or both sides as shown in D. The vertical member may be constructed solid as at E or adjustable as at F.

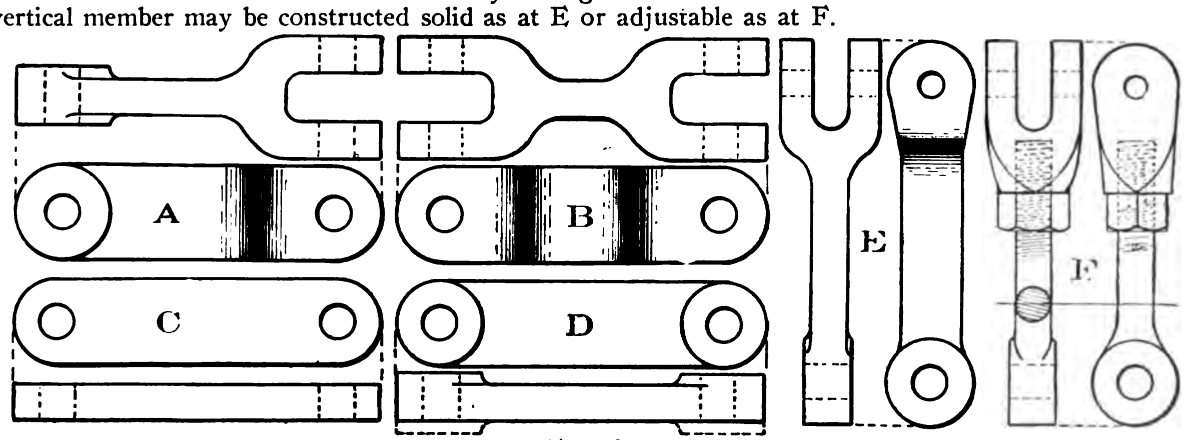


FIG. 69.

Fig. 70 shows some other methods of designing the toggle.

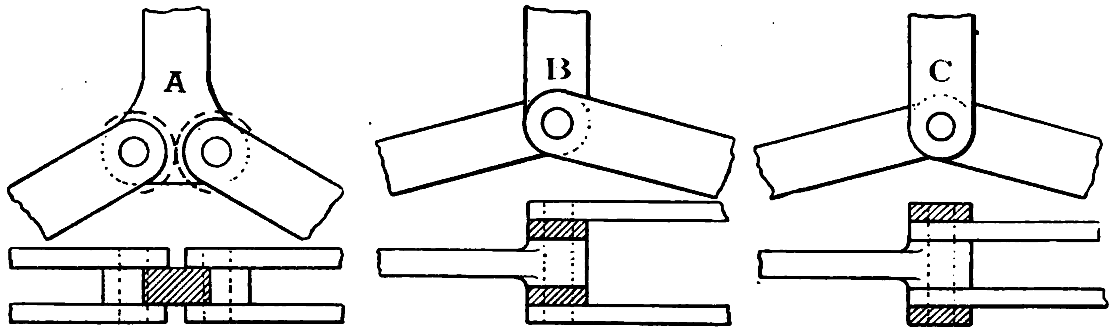


FIG. 70.

165. Die Heads:—The following shapes Fig. 71, may be of value in designing the sliding head and stationary block. *A* shows the simplest arrangement for fastening these heads to the bed. The slid-

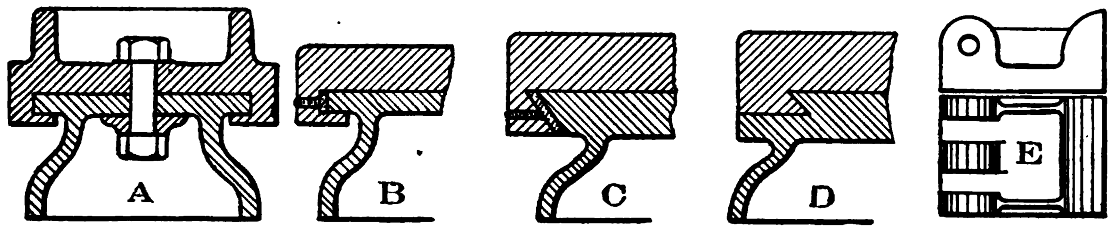


FIG. 71.

ing head would be made as at *E* for fastening to the toggle and the stationary block would be bolted to the bed as shown at *A*. In such a design the overlap below the top of the bed should be made sufficiently strong to resist the turning action from W_3. *B* and *C* show the application of gibs between the sliding head and the bed to take up side slack. In some classes of machines such an arrangement would be considered essential. If, however, heavy side thrusts were involved as in *C* this form would be questionable. With the bed planed to an angle as at *C* and *D*, the latter would be considered the stronger.

166. Sliding Head:—Since the sliding head can not be rigidly fastened to the bed, it must be fitted to a set of guides. The most common fastening is shown in Fig. 72. Having the forces W_3 and W_4 acting on the pin and allowing all the reaction from the die to fall at the upper point of the head we have a cantilever beam acted upon by three forces and tending to break at some section as *a, b*. Taking W_3 in two moments about *a*, and W_4 in direct pressure we have

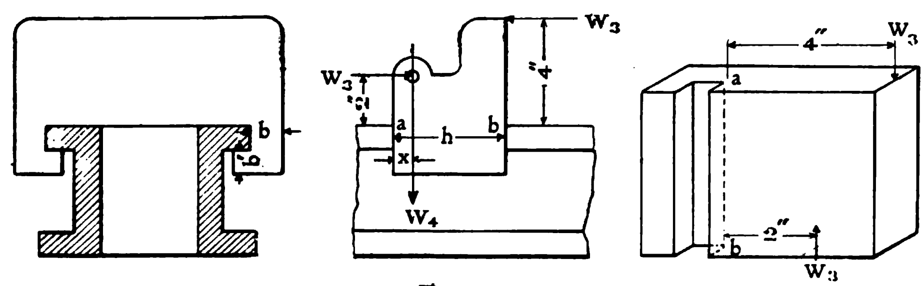


FIG. 72.

$f_p = \frac{450}{2bh}$ due to the pressure; then if $f_m - f_p = f_t$ we have $\frac{15312}{bh^2} - \frac{450}{2bh} = f_t$.

Let $h = 5''$ and $f_t = 1500$ we have $b =$ approx $\frac{3}{8}''$.

If the fibre strength of tension and shear of cast iron be taken the same then $b' = b$.

In like manner the reaction W_3 from the die may be taken at the bottom instead of the top of the sliding head, and the turning moment figured in this way to see if there is greater danger to the section than when taken at the top.

167. Stationary Head:—The fastening of the stationary block to resist the force W_3 in the large machines will call forth extreme care on the part of the designer. The simplest fastening is shown in Fig. 73. Take $W_3 = 2552$; $y = 4''$; $x = 6''$. we have, disregarding the benefit obtained from the overlap of the block around the bed, $2552 \times 4 = 6\ W\ ;\ W\ = 1702\#$.

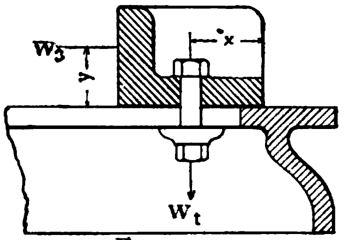


FIG. 73.

It is now necessary to determine if the block will slip with this force binding the two surfaces to the bed. Let the coefficient of friction between the blocks and the bed be Φ = say .3, then the resistance due to the friction will be $1702 \times .3 \times 2 = 1021.2\#$. (Force normal to plane $\times$ coeff. of friction = Resistance parallel to plane). This is about 40 per cent. only, of the amount needed to keep the block from sliding. We should now find out what force will be required in the bolt to just balance the 2552 pounds. Substituting in the formula $W_t\ \Phi \times 2 = W_3$ we have $W_t = \frac{2552}{2 \times .3} = 4253.5$ pounds exerted at the root of the thread tending to stretch the bolt. With $f = 8{,}000$, this will call for approximately .5□″ area, or a 1″ diameter bolt. It is evident from this that some other arrangement should be used than a plain friction surface. In Fig. 74, A is very similar to Fig. 73; the lower block in this case engaging in a series of notches which protect it

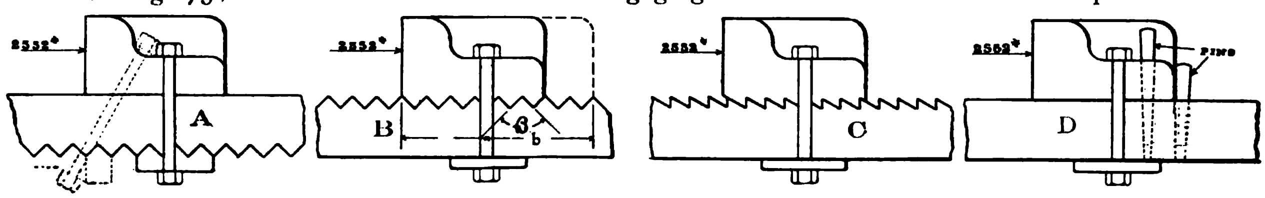


FIG. 74.

against slipping. The upper block may slip slightly, but this will cause a greater grip and a consequent increase of frictional resistance. A possible improvement on this (if the construction of the machine would permit it) would be to have the bolt at an angle as shown in the dotted lines.

Let this angle be, say, 30° with the horizontal, then from Fig. 75, *A*,

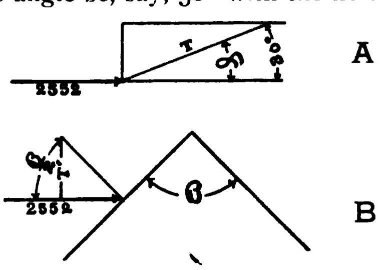


FIG. 75.

$.3\quad T \sin \alpha =$ resistance due to friction.

$T \cos \alpha =$ horizontal effort.

$.866\ T + .3 \times .5\ T = 2552.$

$\therefore\quad T = 2512$ pounds $= 1\frac{3}{16}''$ bolt.

Fig. 74, *B*, will cause a tension on the bolt (disregarding friction) of $T' = 2552 \tan (\beta \div 2.)$ Let $\beta = 90°$ then $T' = 2552$ pounds $= 1\frac{3}{16}''$ bolt, the same as 74 A. It is very evident that if friction were included in this it would reduce the bolt size somewhat below 74 A. C is probably not as strong in the shape of the tooth as *A* and *B*, but with a large tooth area the unit shear becomes small enough so that the teeth are not endangered. The vertical faces on the teeth reduce the vertical thrust on the bolt to a minimum and permit the use of a bolt just sufficiently strong to protect the block from turning as in Fig. 73.

D is arranged to have pins to fasten into the bed either through the block, or behind it. These pins keep the block from sliding and are calculated for shear, while the bolt is calculated for turning as in Fig. 73.

Another way in which these fastenings may give way is by shearing the bolt. Assume W_3 Fig. 73, entirely acting to shear, we have $\frac{2552}{f = \text{say } 5{,}000} = .51$□″ of bolt area. If this is taken as the full area of the bolt it would be $1\frac{3}{16}''$ diameter. This shows a requirement about equal to those for tension. In any form of fastening it is well to investigate both tension and shear and take the larger requirement.

It should be understood also that, if the block clamps over the bed on planed ways, it will assist the bolt in holding the block down and a smaller bolt may be used.

168. Bed:—The calculations for the bed will be found to be more complicated. Assume a simple bed say of the same general shape, and cross section as Fig. 76. Assume also the

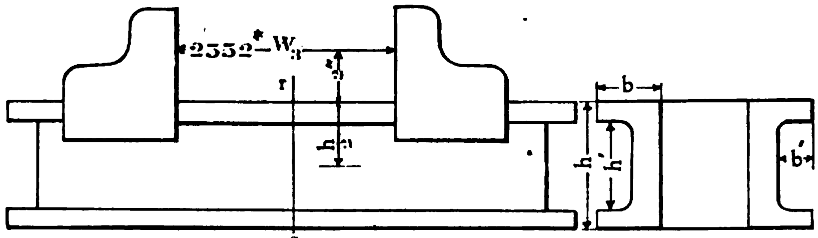


FIG. 76.

force $W_3 = 2552\#$, acting at the middle of the block face a distance of say 2″ above the top of the bed This force W_3 tends to break the bed along some line as $r\,s$ putting a combined bending and tensional stress on the fibres of the section. Considering the part to the right of the section as free we have, Fig. 77, the fibres on the upper or weakside subjected to two tensional stresses the sum of which should not exceed the safe fibre stress of the metal, i. e., $f_1 + f_2 = f_t$; and the fibres on the lower side, subjected to a tensional and a compressional stress, the algebraic sum of which should not exceed the safe compressional fibre stress of the metal, i. e., $f_1 + (-f_2) = f_c$ where

f_1 = uniform tensional stress

f_2 = stress due to bending

f_t and f_c = combined stresses.

To obtain f_1 and f_2 on the tension side take

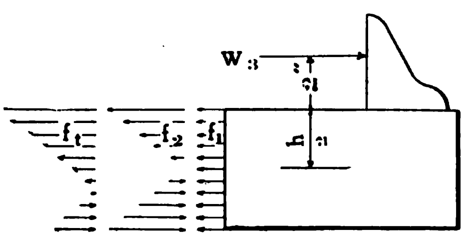


FIG. 77.

(1), $\frac{\quad}{A} = f_1$ where A = area of section in sq. in.

(2), $\frac{W_3\,(h \div 2 + 2)}{Z} = f_2$ where Z = modulus of the section.

Having selected the section of the bed as Fig. 28 we find the modulus to be

$$Z = \frac{b\,h^3 - b'\,h'^3}{6\,h} \times 2$$

It will be necessary here to select some values for b, b', h and h' and make a trial solution. Take $b = 2''$; $b' = 1\tfrac{1}{2}''$; $h = 6''$ and $h' = 4''$.

Substituting these values in the above we have

$$Z = \frac{2 \times 216 - 1.5 \times 64}{6 \times 6} \times 2 = 18.8$$

Substituting this value in (2) we have

$$\frac{2552\ (3 + 2)}{18.8} = 679\#\square'' = f_2$$

With the values of b, b', h and h' as given, the area of the entire section becomes $A = 12\square''$ and

$$\frac{2552}{12} = 212.7\#\square'' = f_1$$

then $f_1 + f_2 = 679 + 212.7 = 891.7\ \#\square'' = f_t$

Since the usual value of f_t for cast iron is about 1500 this shape and size of section would be stronger than necessary.

Now, if the figures of the section be changed to read $b = 2''$; $b' = 1\tfrac{1}{2}''$; $h = 5''$ and $h' = 4''$ the values become

$$f_1 = \frac{2552}{8} = 319\# \qquad\qquad f_2 = \frac{2552\ (2\tfrac{1}{2} + 2)}{10.2} = 1126$$

$f_1 + f_2 = 1126 + 319 = 1445.$

This seems to agree very well with the value of $f_t = 1500$. Since cast iron is much weaker in tension than in compression, the latter will not need to be investigated and the above figures can be accepted for the size of bed.

Having found the shape of the simple section it is possible to modify it to a certain degree without affecting the calculations seriously.

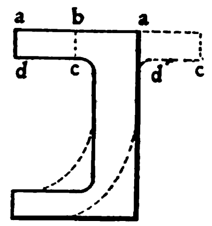


FIG. 78.

To illustrate, the portion $a\ b\ c\ d$ Fig 78 may be lopped off and added to the inner side at $a'\ b'\ c'\ d'$ without affecting the modulus. Fillets may then be added at the interior corners giving a shape similar to most bed tops.

For the bottom, a slight deflection or slope of the web can be made as shown by the dotted lines and the result is very similar to a plain cast iron engine or lathe bed. Other minor changes such as slight curves instead of straight sides might be made without any loss of rigidity. In any case where the shape of the simple section is found and the designer wishes to increase the thickness of any part he may do so and the result is merely to increase the factor of safety.

If under very heavy loads it is advisable to specify one or more steel I beams or channels from Cambria, this may be done by making a trial selection of a section and substituting the values of Z and A in the formulas as before. If this value $f_1 + f_2 = f_t = 16{,}000$ the condition is fulfilled as in the case of the cast beam.

169. The final determination to be made in this design is to obtain the length of the bed to prevent overturning when the load is applied. Let W_5 Fig. 79 = the weight of the bed, then from the force diagram we have the following moments about the end at b.

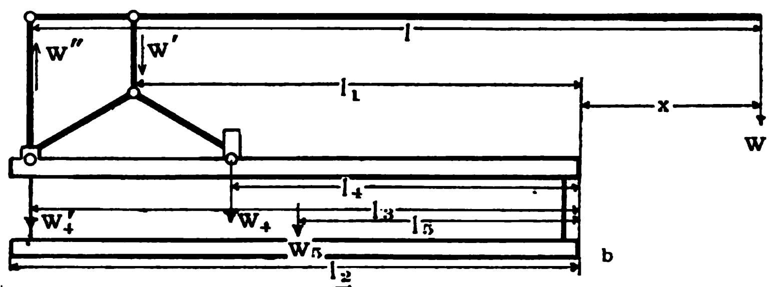


FIG. 79.

$$W\,x + W''l_3 - (W'_4\, l_3 + W_4\, l_4) - W_5\, l_5 = 0\ .\ .\ .\ . \tag{1}$$

but $\quad W'_4\, l_3 + W_4\, l_4) = W''\, l_3$ and (1) becomes $W\,x = W_5\, l_5$ (2)

The length of the bed may be obtained from (2) by adjusting the values of x and l_5 such that the equation will be satisfied.

To obtain the length however, in a more direct way the following can be used:

If $x = l - l_3$ and $l_5 = l_2 \div 2$ then (2) becomes $W\,(l - l_3) = W_5\, l_2 \div 2$ (3)

Knowing the cross section of the bed in square inches, the weight of one inch in length would be $.26\,A$; the total weight of the bed being $.26 l_2\,A$. From (3) we get $l_2^2 = W\,(l - l_3) \div .13\,A$.

Let $l_3 = l_2 - a$ where a is the offset as shown, then $l_2^2 = W\,(l - l_2 + a) \div .13\,A$, from which is obtained the formula

$$l_2 = -\,3.85\,\frac{W}{A} \pm \sqrt{7.7\,(l + a)\frac{W}{A} + 14.82\left(\frac{W}{A}\right)^2} \tag{4}$$

In the above the factors W, A, l and a are known and by substitution we get the length of the bed to just balance this force W. Any amount more than l_2 as found will insure not overturning.

CHAPTER VI.

DESIGN NO. 2.

170. A Belt Driven Punch or Shear is the machine selected to represent the second general design. Included within this one machine are problems covering the design of frame, levers, gears, fly wheel, pulleys, bearings, shafts, sliding head, punch, die, clutch, stripper and cam. The fact that this machine finds such general use in manufacturing plants and that it embodies such a variety of designs makes it an ideal subject for analysis. Fig. 80 shows a motor driven shear of the latest design. It is not expected that the following design will be for a motor drive but that the distance between the bearings be shortened and pulleys used instead. In giving out the design the following requirements will be made: First, the work to be accomplished, i. e., diameter and depth of hole to be punched or the cross section of the piece to be sheared; second, the distance from the edge of the plate to the center of the hole, or the depth of the throat of the machine; third, the average cutting velocity of the punch or knife in inches per second, or the R. P. M. of the cam shaft.

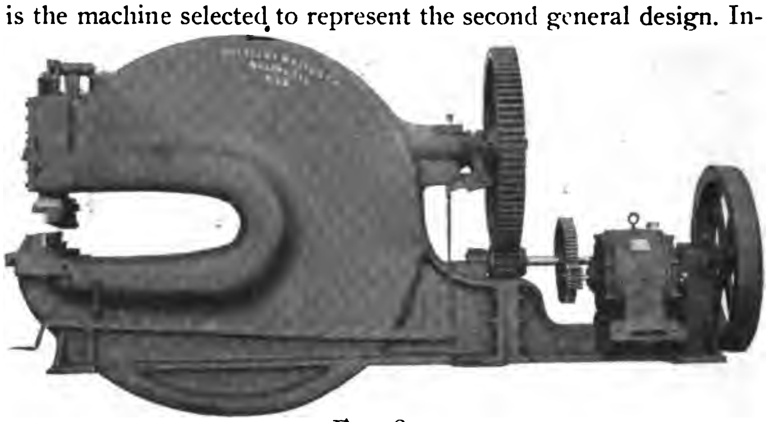


FIG. 80.

In the analysis of the methods employed in working up such a design, the frame sections will be carried out somewhat in detail because of the advanced character of the work; the rest of the machine will be dealt with more briefly. Reference will be made to Part I for formulas relating to tension, compression and shear. In making the assignments, the members of the class will be given values that will differ materially from those worked out here. The three preceding pages show a sample set of drawings of such a machine.

171. Required in the Design:—A machine to punch a one (1) inch hole through three quarters ($\frac{3}{4}$) inch mild steel plate, the center of the hole to be not greater than seven (7) inches from the edge of the plate. The velocity of the punch during cutting may be taken in this case as approximately one (1) inch per second.

172. Frame:—The material used in the frame of such a machine is either close grained cast iron or cast steel. The general shape is about as shown in Fig. 84 and the sections of the frame Fig. 81 are either hollow cast iron as shown in B and C or web shaped steel as shown in A. Of the three sections, B and C are the most common.

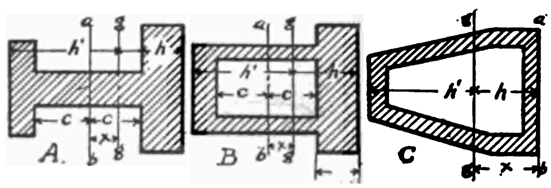


FIG. 81.

Let Fig. 87 represent the outline of the assembly drawing and x, x be the center line of the frame. On this line lay off the throat depth, say 8 inches. Select the shape of the horizontal section, say B, from the standard forms and work out the size of the section by the method employed in the Bed, par. 168. Taking G as the depth of the throat and h as the distance from the edge of the casting to the center of gravity of the section. In applying the formula $f_2 = \frac{W\,(h + G)}{Z}$ there may be some confusion in obtaining a satisfactory value for Z owing to the unsymmetrical section. To get Z it will be necessary to determine the *moment of inertia* I of the section and then find Z by the following:

$$\text{for tension } Z = \frac{I}{h} \qquad\qquad \text{for compression } Z = \frac{I}{h'}$$

Make a trial selection of some sizes for the section and find the neutral axis by cutting out a pasteboard section and balancing it on knife edges, or by the following: assume any line of reference as $a\,b$

Fig 81, take the algebraic sum of the moments of each rectangular section about this line of reference and divide by the total area; this will give the distance x between the line of reference and the neutral axis $g\ g$ of the section. When $g\ g$ is determined find I by the following: To the sum of the products of each area by the square of its distance from the neutral axis add the moment of inertia of each section about its own neutral axis. It will be remembered that the moment of inertia of any rectangle about its own neutral axis is $I = b\,d^3 \div 12$ where d = the total height of the section.

Assume the section with sizes as shown in Fig. 82, then

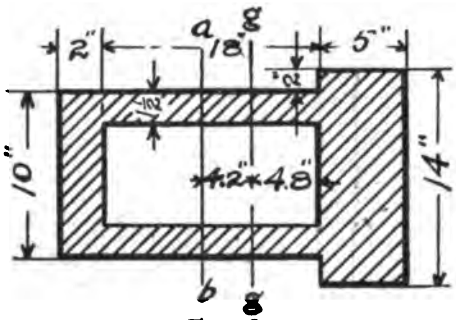


FIG. 82.

$$x = \frac{70 \times 11.5 - 2 \times 10 \times 10}{70 + 20 + 54} = 4.2''$$

$$I = 70 \times (7.3)^2 + 54 \times (4.2)^2 + 20 \times (14.2)^2 +$$

$$\frac{14 \times (5)^3}{12} + \frac{3 \times (18)^3}{12} + \frac{10 \times (2)^3}{12} = 10326.$$

$$Z = \frac{10326}{9.8} = 1054 \text{ for tension}$$

$$Z = \frac{10326}{15.2} = 679 \text{ for compression.}$$

The value W in the formula is the load on the punch and in this case, if the ultimate shearing stress of mild steel be taken at 55000 pounds per square inch, it would be 129,591 pounds.

Considering the section only on the tension side we have $f_1 + f_2 = f_t = 900 + 2189 = 3089$#□″. This fibre stress would be large for cast iron hence another section must be selected.

Take for a second trial the section Fig. 83, we have if worked as above

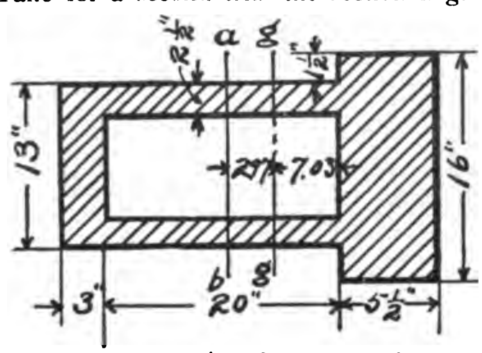


FIG. 83.

$x = 2.97''$

$I = 21049.44$

$$Z = \begin{cases} 1680 \text{ for tension} \\ 1317 \text{ for compression.} \end{cases}$$

$f_1 + f_2 = f_t = 2154$ #□″

In like manner we may work out the compression side by $f_1 - f_2 = f_c$.

Any other section of the frame can be determined by working out f_2 as shown above and combining with it the value of $f_1 = W \cos \alpha \div$ area. The value of f_1 is a maximum when α is zero and becomes zero when α is $90°$. It will be seen, Fig. 84, that at section A, $f_1 = W \div$ area A; at B, $f_1 = F_b \div$ area B, but $F_b = W \cos \alpha$, hence $f_1 = W \cos \alpha \div$ area B; at C, $f = W \cos \alpha \div$ area C and so on until f_1 becomes zero at section E. At this point the frame should be examined for bending and shearing and the larger requirement taken. In all probability section E will be made larger than the calculated size to accommodate the finishing around the head. It will be satisfactory in this design if we obtain sections at $\alpha = 0°$, $45°$ and $90°$.

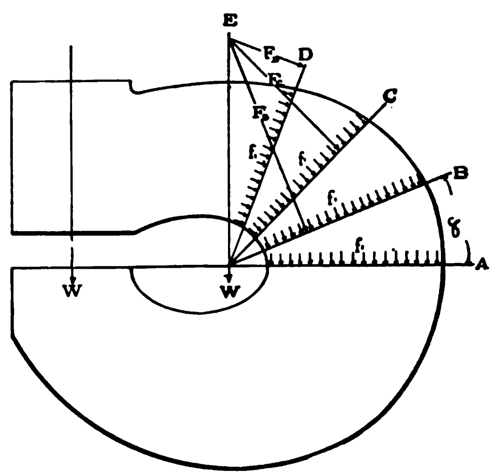


FIG. 84.

To find a section at say $\alpha = 45°$ determine the height of the gap from the character of the work to be performed and draw the outline. Assuming

some section of the frame as Fig. 85, and solving for the fibre stress as before, we find

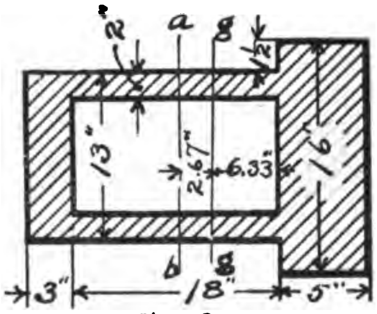


FIG. 85.

$$x = 2{,}67''$$

$$I = 15655.$$

$$Z_t = 1382.$$

$$f_1 + f_2 = 1872 \text{ \#□}''.$$

NOTE.—In finding M in $M = f\,Z$ the lever arm varies approximately with the cosine of the angle a.

For the vertical section take Fig. 86, in which case we have,

$$x = 3.26''$$

$$I = 4411.79$$

$$Z_t = 655.$$

$f_2 = 791$ which shows that the section could be materially reduced if it were desired. This reduction could very properly be made according to the dotted lines.

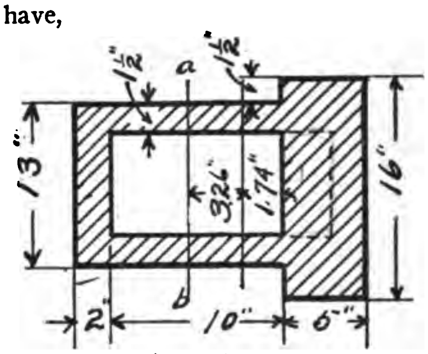


FIG. 86.

To investigate for shearing on the vertical section we have, allowing the shear to be absorbed by the entire section:

$$f_s = \frac{129591}{136} = 953 \text{ \#□}''$$

173. Having determined several important sections in the frame, the outline can be drawn in on Fig. 87. This outline will of course be modified somewhat for the shaft, head and leg.

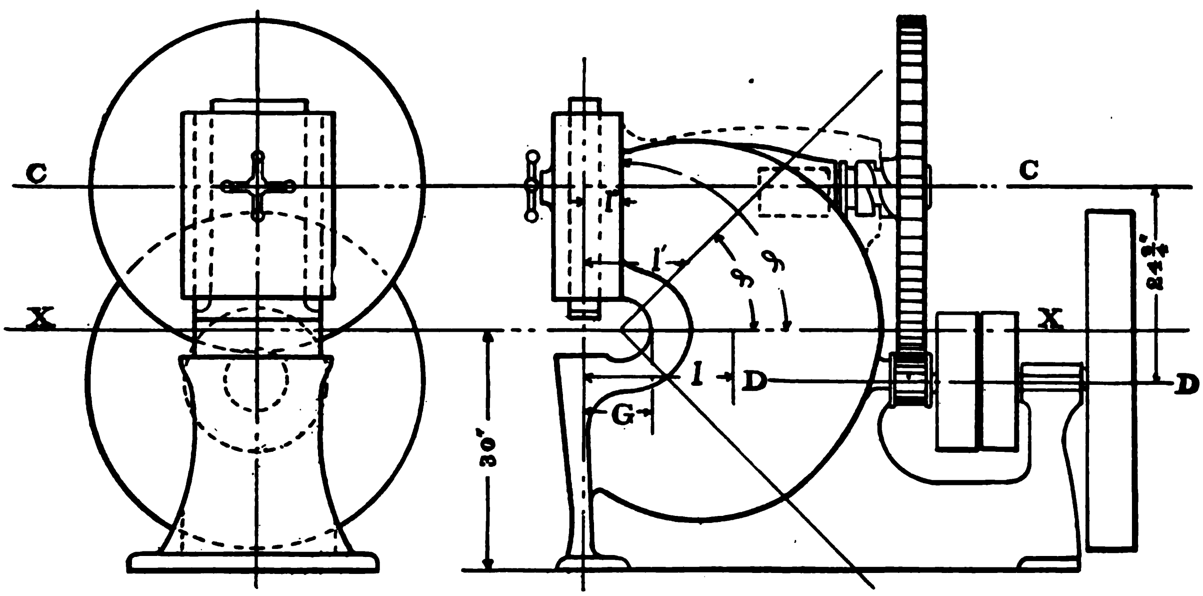


FIG. 87.

It will be noticed that a somewhat larger fibre stress has been allowed in this frame than in the material as used in the toggle frame. This is about as would be expected; any casting planned to fill a very important place in the design of any machine would be made of the best close grained gray iron. It is advisable to keep the size of this frame as small as possible consistent with strength and since the best of cast iron would have an ultimate strength of 25,000 to 30,000 pounds it would be considered safe in this to allow a fibre stress of 2,000 to 2,500 pounds per square inch, which corresponds to a factor of safety of 12.

The shape of the section may be varied to suit the conditions, from a large and thin section as here treated, to a small compact and possibly solid section. The latter condition prevails in some machines where the gap is long and the main section would be necessarily crowded into the smallest space.

Cast steel frames are very common, especially on the larger machines. When made of steel the frame section may be made much smaller. f_t = 12000 to 15000 #□″.

Tension bars are provided for machines with long gaps. These bars are very necessary when doing heavy duty.

174. Shaft:—The shaft which would be planned according to Figs. 88 or 89 would be made of hammered steel. B_1 B_2 and B_3 are journals; C is the cam which operates the punch. The greater part of the thrust from the punch is absorbed at the journal B_2, B_3 being added chiefly to reduce the strain due to bending.

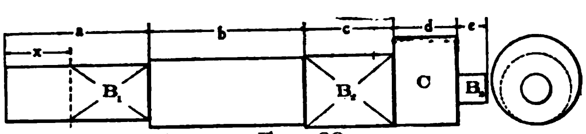


FIG. 88.

In designing the shaft the diameter of the part *a* should be figured to resist the twisting moment from the gear *x*, allowing the fibre stress of about 6000 #□″ for shear. It will be noticed that the rotating force also produces a bending moment on the shaft, this bending moment however, is so small compared to the twisting moment that by taking alow fibre stress as above, it may under ordinary conditions be neglected. The length of the journal may be taken from 2 diameters to 2.5 diameters of the shaft. The length of *b* will be quite variable and will be governed by the frame of the machine. The diameter of *b* will depend upon the judgment of the designer; in some shafts it is made equal to the diameter of the left journal while in others it is enlarged to the size of the main journal. A high speed machine would require a larger shaft than a slow speed machine, because of the heavy shocks to which the shaft is subjected.

The diameters of the main journal and the cam will be correlative. Take the size of the cam such that the pressure per square inch of projected area will not exceed 8000 pounds; say 5000 pounds for Fig. 88. The length of the cam will vary for ordinary sizes of machines from 3 inches to 6 inches and the diameter from 5 inches to 10 inches. When the cam is planned, make the main journal flush with the near side of the cam. The diameter of B_2 then will be governed by the eccentricity of the cam. The length of the main journal should be such as to give not more than 3000 pounds per square inch of projected area, assuming the entire thrust to be absorbed at this journal.

In some machines it is desired to have the journal B_2 as short as possible and have the diameter larger than the cam. This can be done as shown in Fig. 89. This form of shaft should have the cam figured for about 8000 pounds instead of 5000 pounds as before.

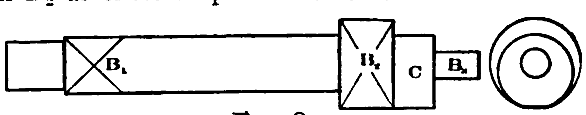


FIG. 89.

The frame of the machine should be fitted with a phosphor-bronze bushing ¼ inch to ⅜ inch in thickness surrounding the journal B_2. This bushing is made a forced fit with the frame.

The size of B_3 would vary between 2 inches and 4 inches for both diameter and length.

In any machine of this kind it is safe to allow about 15 per cent for the friction of the parts while performing the heaviest duty. The total pressure to be accounted for at the driving end will then be $129591 \div .85 = 152460$ pounds. If the eccentricity of the cam be taken the same as the thickness of the thickest metal to be punched = ¾ inches, the twisting moment from the gear side will be approximately $\frac{3}{4} \times 152460 = 114345$ ″#. Applying this to the formula for twisting, par. 60, it gives d = 4.59 inches. Calling this 4.5″, it represents the diameter of the shaft at the smallest section.

The cam diameter, assuming a length of 4" and a pressure per square inch of 5000 pounds is
$\frac{129591}{5000 \times 4} = 6.5$ inches.

B_2 will then be 5" diameter and if we allow 2500 #□" pressure will have a length of $\frac{129591}{2500 \times 5} =$ 10.4 inches, say 11 inches.

B_3 may be taken say 2½ inches long by 3 inches diameter.

175. Sliding Head:—Of the different forms in use, two of the very common ones are shown in Figs. 90 and 91, the former being more common in the smaller types of machines. The chief objection to this form is its liability to wear unevenly thus causing lost motion and an irregular movement of the block while punching.

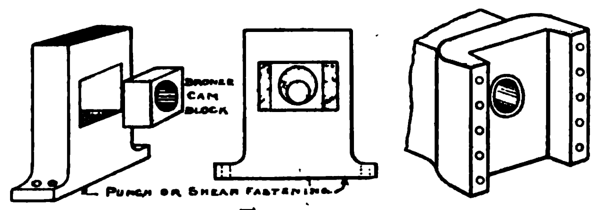


FIG. 90.

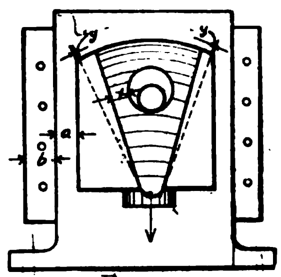


FIG. 91.

In the latter form, the entire thrust is carried on a steel block set into the sliding head and the wear, if any, is practically uniform. The size of the bearing surface in the steel block may be taken from the crushing strength of the steel. If this value be taken for the best carbon steel at 150,000 pounds with a factor of safety of 10, the projected area of this bearing will be $\frac{129591}{15000} = 8.6$ square inches, from which if the length of the cam be 4 inches, the breadth of the bearing will be 2.15 inches say 2¼ inches. The breadth of the sliding head will be seen to depend upon the construction of the vibrating arm. Make the vibrating arm of cast steel and allow from ¾ inch to 1¼ inches at x and a small clearance at y. The values a and b will depend upon the width of the frame section.

176. After planning the sliding head locate approximately the center line of the cam shaft as c.c. Fig. 87. In locating the pulley shaft D D it will be necessary to first find the diameter of the gears. Since the R. P. M. of the small gear is the same as that of the pulleys this value may be taken from the following table, No. XIX, which represents current practice.

TABLE XIX.

Machine will Punch	Diameter of Pulley.	R. P. M.
¼" x ¼"	10	200 to 250
½" x ½"	12	200 to 250
¾" x ¾"	16	175 to 200
1" x 1"	18	150 to 175
2" x 1"	30	150 to 175

Knowing the angular velocities of the two shafts the diameter of the small gear may be assumed and the distance between the shaft centers obtained. In this machine if the cutting speed of the punch is one inch per second, the center of the cam will travel approximately $60 \div 4.71 = 13$ R. P. M. Calling this 15 R. P. M. and the R. P. M. of the pulley shaft 150 the ratio of the gears is 10. With 4½ inches as the diameter of the pinion, the shafts will be 24¾ inches between centers.

NOTE.—The value 4½ as the diameter of the pinion was taken merely for illustration. This would be rather small for the construction of a perfect tooth.

177. Fly Wheel Weight:—In determining the weight of the fly wheel for a machine of this kind the results may vary between wide limits from a wheel such that its kinetic energy will just equal the energy absorbed by the machine during punching, in which case if we disregard the belt's action, the velocity of the wheel would become zero after each hole punched, to a wheel of such a size that the residual energy will be sufficient to keep the speed fairly constant. Current practice approaches the former and in this consideration will be adopted. Having given the force to be accounted for at the

driving end as 152460 pounds, assume that this force acts through say a maximum of ½ the total depth of the cut which is ⅜ inch or $\frac{1}{32}$ foot, the energy exerted would be 4764 foot pounds. Take the formula $W v^2 \div 2g$ = Kinetic energy of any moving body and apply the formula to the mean rim diameter. Assuming 36 inches as this diameter we have:

$$\frac{W v^2}{2 g} = 4764; \quad W = 553 \text{ pounds.}$$

178. Arm:—The fly wheel arm may be calculated as follows: Estimate the time required in punching one hole then find the distance through which a point on the center line of the rim will move during this time; this will be the value V in P V = 4764. Since there are 15 R. P. M., each revolution will take 4 seconds. Assuming the velocity of the punch during action to be the same as that of the cam center we have $V = 3.1416 \times 1.5 \div 4 = 1.1781$ inches per second. The time occupied in punching is $\frac{3}{8} \div 1.1781 = .318$ second. The velocity of the rim of the wheel is 1413.7 F. P. M. = 23.56 F. P. S., from which we find that the rim will travel 7.5 feet before stopping. Applying P V = 4764

$$P = 635 \text{ pounds.}$$

The value *P* may be found in another way. Find the force *p* at the gear, Fig. 92; from the moments around the cam shaft, this is

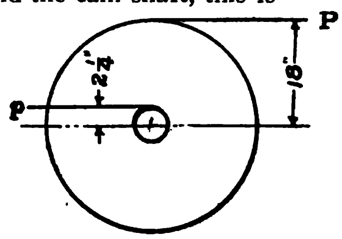


FIG. 92.

$$p = \frac{152460 \times 3}{4 \times 22.5} = 5082 \text{ pounds.}$$

then by moments around the driving shaft

$$P = \frac{5082 \times 9}{4 \times 18} = 635 \text{ pounds.}$$

Having found *P*, obtain the large dimension of the arm at the center of the shaft from formula par. 107.

$$b = \sqrt[3]{\frac{P R}{6 \times .05 \times 1000}} = \sqrt[3]{\frac{635 \times 18}{6 \times .05 \times 1000}} = 3.37'' = 3\tfrac{3}{8}''$$

For details of shapes and sizes of rim, arms and hub see par. 113.

179. Working Depth of the Cut:—The actual cutting depth of a punch or flat shear is used in determining the foot pounds of work done at the tool, and is a certain percentage of the total thickness of the metal. This percentage varies somewhat with the kind of the metal, but for mild steel it has been found by experiment (American Machinist, Oct. 12, 1905) to be

Thickness of Metal in inches	1	¾	⅝	½	⅜	$\frac{5}{16}$	¼	$\frac{3}{16}$	⅛	$\frac{3}{32}$	$\frac{1}{16}$	$\frac{1}{32}$
Penetration in per cent.......	.25	.31	.34	.37	.44	.47	.5	.56	.62	.67	.75	.87

180. The maximum punching or shearing force is used in calculating the frame sections. The ultimate shearing stress of the metal to be cut multiplied by the area gives the maximum load on the punch or flat shear. If the greatest load on a *bevel shear* is desired, multiply the maximum load on a flat shear by the following:

	THICKNESS OF THE METAL.										
	[illegible]	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2
4° Bev	.42	.48	.54	.61	.67	.73	.79	.85	.92	.98	
8° Bev	.23	.3	.37	.44	.51	.58	.65	.73	.81	.88	.95

Look up articles on the Shearing of Metals in the American Engineer and Railway Journal. Vol. 67, Page 142.

181. Driving Shaft:—This shaft will be subjected to both bending and twisting. With the bearings close up to the pulley and gear the bending will not be great and the shaft may be designed to resist twisting with *f* taken at 6000 the same as in the cam shaft. Worked out in this way the diameter becomes 2.2 inches say 2¼ inches.

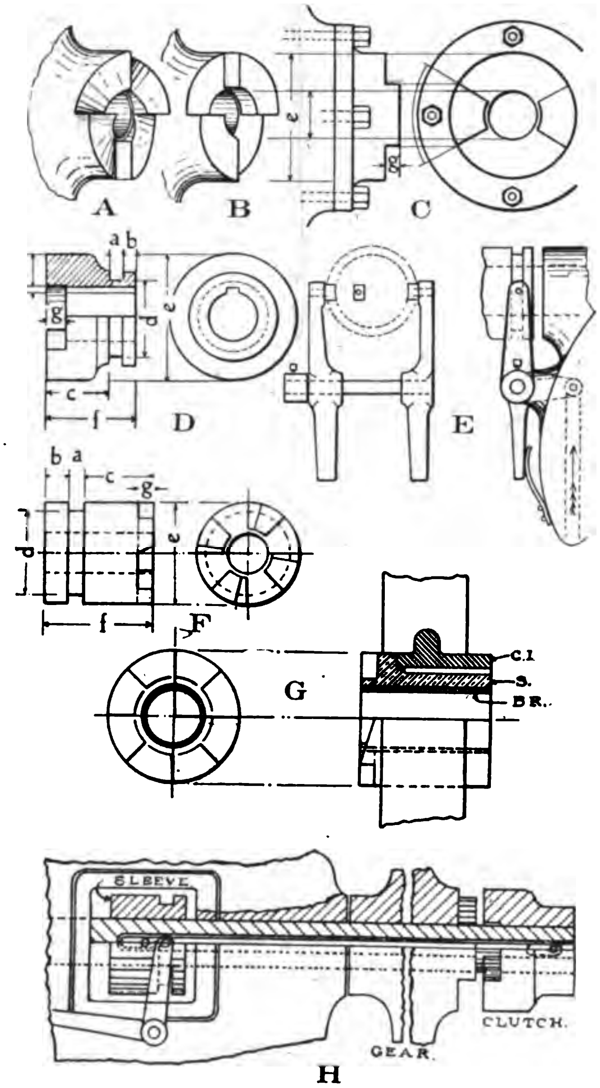


FIG. 93.

182. Pulleys:—To determine the width of belt and pulleys to use, no definite rule can be stated. Current practice ranges from a 2 inch belt on a ¼" x ¼" machine to a 6 inch or 7 inch belt on a 2" x 1" machine. It is suggested however that the diameter of the wheel be selected and the effective pull P on the belt obtained assuming the punch to be acting all the time, then multiply this by the per cent of time the punch is actually in service and figure the belt from this result.

The belt sizes obtained in this way will in all probability not be very satisfactory and the suggestion is made merely to stimulate investigation. Satisfactory sizes can be obtained from catalogs as a final resort.

For sizes and shapes of pulley parts see par. 115.

183. Clutches:—In operating any machine having an intermittent motion a clutch is commonly used to serve as a connector between the power supply and the work. The application of the clutch to the simple punching or shearing machine is shown in Fig. 93. It is usually applied directly to the hub of the large gear and is operated through a system of levers and cranks by either hand or foot. When the punch is not operating, the large gear which is designed with a long hub to act as a bearing runs loose, the shaft remaining stationary. The clutch sleeve slides on the shaft over a splined key and when the punch is to be operated this sleeve is thrown to engage with the corresponding part on the gear hub. When the hole is punched a counter weight brings the sleeve back to its former position and the movement of the punch ceases.

Clutches are formed with either two, three or four jaws. These jaws may be formed as a part of the wheel hub as shown at A and B, cast from steel and bolted to the flat face of the wheel hub as shown at C, or cast from steel and fitted to the interior of the wheel hub as shown at G. In heavy work C and G are preferable.

The part of the clutch subjected to the greatest wear is the front face of the jaw. This is sometimes faced with a plate of high carbon steel which can be replaced when necessary with a fresh one. The rear face of the jaw is usually perpendicular to the front face of the wheel but is sometimes cut to an angle of 30 to 45 degrees. There should be sufficient clearance between the jaws on the sleeve and the wheel to enable them to be easily thrown together while in motion. This should be from ⅛ inch to ¼ inch.

The clutch sleeve may be shaped as shown in either D or F. The following sizes table XX, will meet average requirements.

TABLE XX.

Shaft =	2"	3"	4"	5"	6
a	¾	1	1¼	1½	1¾
b	¾	1	1¼	1½	1¾
c	=	f	—	(a + b)	
d	4	5½	7	9	10½
e	5	7	9	11	12
f	3	4	5½	7	8
g	¾	1	1¼	1⅜	1½

There are two general methods of designing the transmission device; the first and simpler one E, having the clutch between the gear and the frame, and the second H, having the gear between the clutch and the frame. The latter method necessitates a hollow shaft in order to obtain a rigid connection between the sleeve and the clutch and is not much used on small machines.

184. Punch, Die, and Holders:—In all punch and die work the die is made some larger than the punch for clearance. The action of the punch on the material is shown in A, Fig. 94; the hole being taper-

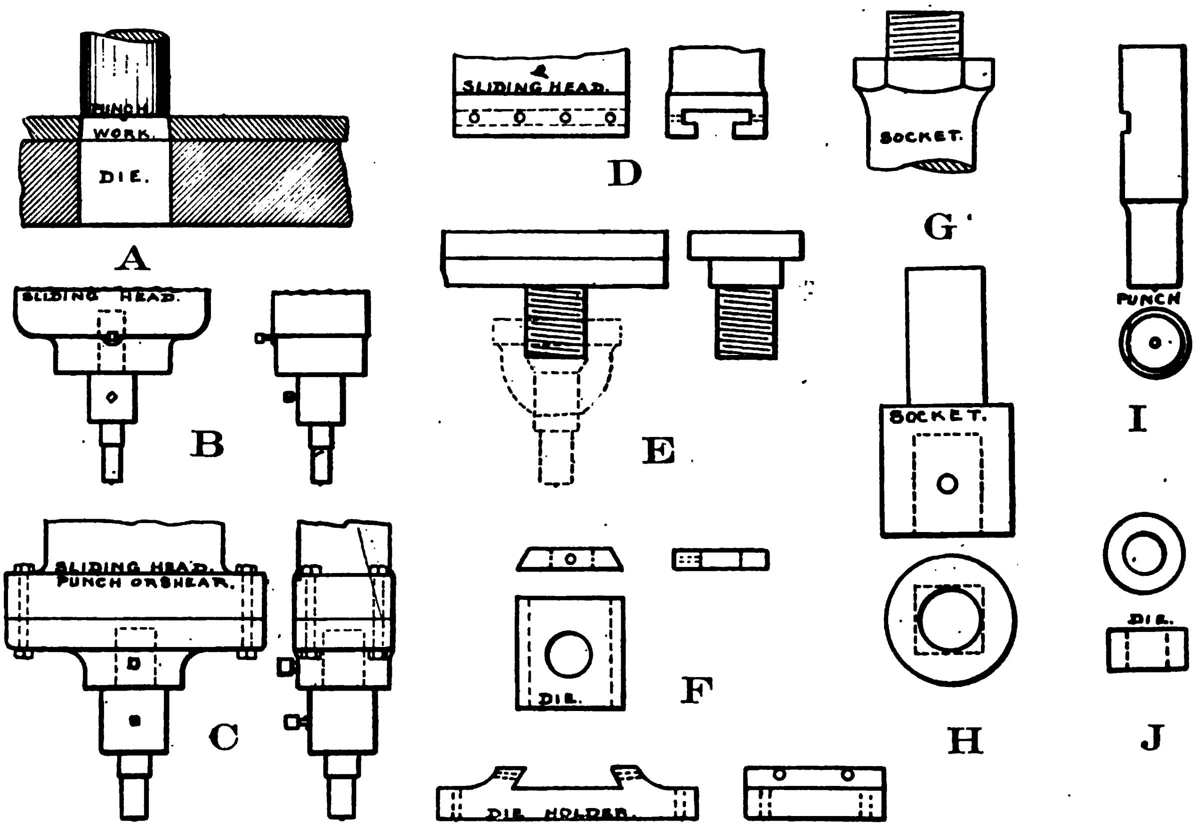
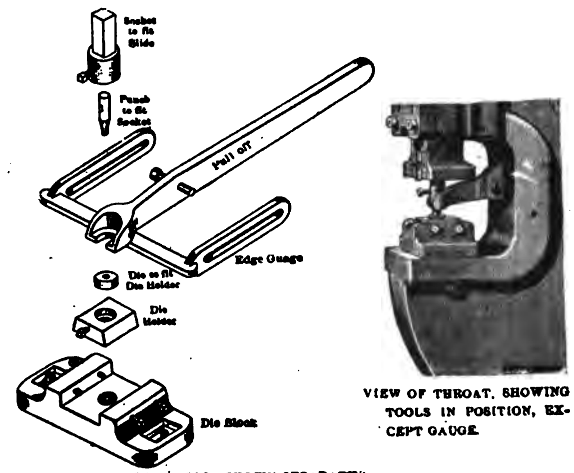


Fig. 94.

ing, having the size of the punch on one side and the size of the die on the other. This taper is slight and is considered of no consequence in rough work but in finished work it is a difficulty that can only be remedied by reaming the hole afterwards.

There are various methods of fastening the punch to the sliding head; *B* shows the bottom of the sliding head fitted with the square ended socket *H* and punch *I*, each held in place by set screws. The screw ended socket *G* may be used equally well instead of the square shanked socket *H*. *C* shows the bottom of the head flanged and drilled for the attachment of either punches or shears. In single machines it is desirable that both punching and shearing be done; where such is the case this is a good form. Side adjustment of the punch may be easily made if the head be slotted as at *D* and fitted with a tee block as *E*. Dies are made from high carbon steel and are held in a holder such as *F*; the holder is in turn bolted to the horizontal face of the frame. A certain amount of adjustment is necesary in locating the die consequently the holder is made in two parts.

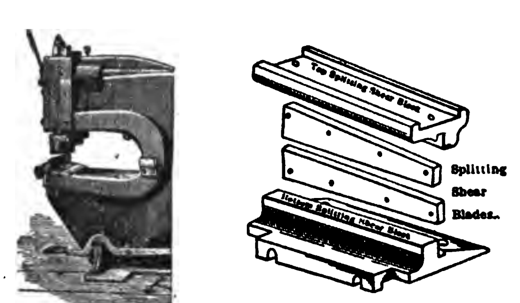
The application of punches and shears to machines is well shown in Fig. 95.



VIEW OF THROAT, SHOWING TOOLS IN POSITION, EXCEPT GAUGE.

A SET OF PUNCHING TOOLS, SHOWN SEPARATED.

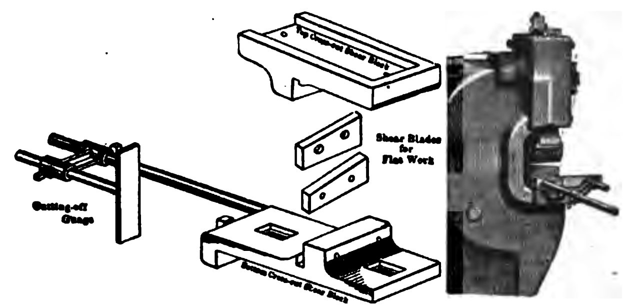
PUNCHING TOOLS.



VIEW OF THROAT, SHOWING TOOLS IN POSITION.

SET OF SPLITTING SHEAR TOOLS, SHOWN SEPARATE.

SPLITTING SHEAR TOOLS.



SET OF CROSS-CUT SHEAR TOOLS. SHOWN SEPARATED.

VIEW OF THROAT SHOWING TOOLS IN POSITION.

CROSS-CUT SHEARING TOOLS.

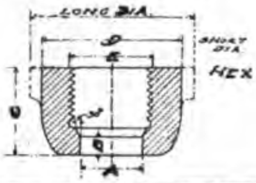
FIG. 95.

The type of punching tool in most common use is that shown in the following figure. The sizes of the various parts are obtained as a matter of experience rather than from calculation. In order that each designer may have quick reference to accurate data on this part the following tables kindly furnished by Messrs. Williams, White and Co., Moline, Ills. are added. In referring to these tables it must be understood that all punches included under one letter are the same length and fit the same coupling nut. It should also be stated that reducer couplings are sometimes provided so that the same stem may be used for different sized punches.

The smallest sizes of punching and shearing machines are operated by hand power or foot power; medium sized machines are operated almost exclusively by belt and the largest machines are operated by belt, steam, water or electricity as shown in Figs. 97, 98, 99 and 100 respectively. These designs show present practice and are added to enable the designer to become more familiar with the form of the parts and the make up of the machines in general.

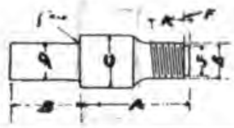
It will be noticed that in the larger machines the frame is of such a size as to project below the floor, the weight being carried on legs or lugs cast on the side of the frame. It will also be noticed that arrangements are made at the top of the frame for the attachment of a crane to assist in handling the material.

COUPLING NUTS



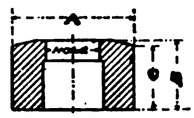
No. of Coupling.	A.	B.	C.	D.	E.	No. of Thr'ds Per In.	No. of Coupling.	A.	B.	C.	D.	E.	No. of Thr'ds Per In.
B	7/16"	1/4"	1"	1 1/8"	1 1/8"	12	H	1 3/4"	7/16"	2"	3 1/2"	2 1/4"	12
C	5/8"	5/16"	1 3/8"	1 7/16"	7/8"	12	I	2 1/4"	9/16"	2 1/4"	3 3/4"	2 3/4"	12
D	13/16"	5/16"	1 1/2"	1 1/8"	1 1/4"	12	J	2 5/8"	9/16"	2 3/4"	4 1/4"	3 1/4"	12
E	1 1/16"	3/8"	1 5/8"	2 1/4"	1 1/2"	12	K	3 1/4"	11/16"	2 3/4"	5"	3 7/8"	10
F	1 1/4"	3/8"	1 3/4"	2 1/2"	1 11/16"	12	L	4 1/8"	1 1/8"	3"	6 1/2"	4 7/8"	10
G	1 7/16"	3/8"	1 7/8"	2 3/4"	2"	12							

STEMS.



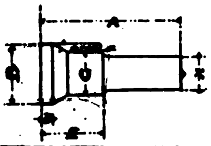
No. of Coupling.	Diameter of Punch.	Regular L'ngth Punch.	A.	B.	C.	D.	E.	F.	J.	K.
C	1/4"– 5/8"..	1 5/16"	1 1/2"	2"	1 5/16"	15/16"	3/8"	5/16"	.731"	5/8"
C	1/4"– 3/8"..	1 5/16"	2 7/16"	2"	1 7/16"	11/16"	3/8"	5/16"	.731"	5/8"
C	1/4"– 5/8"..	1 5/16"	2"	2 1/8"	1 7/16"	1 3/16"	3/8"	7/16"	.731"	5/8"
D	5/16"– 5/8"..	1 7/8"	2 7/16"	2 3/8"	1 5/16"	1 3/16"	1 1/8"	5/16"	1.10"	1 1/16"
E	11/16"– 7/8"..	2 1/8"	2 5/8"	3"	1 5/16"	1 5/16"	1 1/8"	5/16"	1.4"	1 1/8"
E	11/16"– 1"..	2 1/4"	2 7/8"	3"	1 15/16"	1 13/16"	1 1/8"	5/16"	1.4"	1 1/8"
F	7/8"–1 1/8"..	2 1/4"	1 3/4"	3 1/8"	1 15/16"	1 13/16"	1 11/16"	3/8"	1.6"	1 1/4"
H	1 1/16"–1 3/8"..	3 1/8"	1 3/8"	3 1/8"	2 1/4"	2 5/16"	2 1/4"	3/8"	1.96"	1 5/16"
I	1 3/8"– 2"..	3 3/8"	1 3/8"	3 3/8"	2 3/8"	2 5/16"	2 3/8"	3/8"	2.42"	1 5/16"
I	1 3/8"–2 1/4"..	3 3/8"	1 3/8"	3 3/8"	2 1/8"	2 5/16"	2 3/8"	3/8"	2.6"	1 5/16"
J	1 13/16"–2 3/8"..	4"	2 1/8"	4"	3 1/4"	2 13/16"	3 1/4"	3/8"	3.1"	1 3/16"

DIES.



Diameter of Punch.	Hole in Die.	A.	B.	C.	Diameter of Punch.	Hole in Die.	A.	B.	C.
1/8" = .125"	5/32"	1 3/16"	3/8"	13/16"	1/4" = .25" ..	9/32"	......	......	......
5/32" = .156"	11/64"	......	......	......	9/32" = .28" ..	5/16"	......	......	......
3/16" = .187"	7/32"	......	......	......	5/16" = .31" ..	11/32"	......	......	......
7/32" = .218"	15/64"	......	......	......	11/32" = .34" ..	3/8"	......	......	......
3/8" = .37" .	13/32"	......	......	......	1 5/8" = 1.62" .	1 21/32"	......	......	......
13/32" = .40" .	7/16"	......	......	......	1 11/16" = 1.68" .	1 23/32"	......	......	......
7/16" = .43" .	15/32"	......	......	......	1 3/4" = 1.75" .	1 25/32"	3 1/2"	1 1/2"	1 11/16"
15/32" = .46" .	1/2"	......	......	......	1 13/16" = 1.81" .	1 27/32"	......	......	......
1/2" = .5" ..	17/32"	1 13/16"	1 1/16"	1"	1 7/8" = 1.87" .	1 29/32"	......	......	......
17/32" = .53" .	19/32"	......	......	......	1 15/16" = 1.94" .	2 1/32"	......	......	......
9/16" = .56" .	19/32"	......	......	......	2" = 2." ...	2 1/32"	4 1/8"	2 1/8"	2 1/16"
19/32" = .59" .	21/32"	......	......	......	2 1/16" = 2.06" .	2 3/32"	......	......	......
5/8" = .62" .	21/32"	......	......	......	2 1/8" = 2.125"	2 5/32"	......	......	......
21/32" = .66" .	23/32"	......	......	......	2 3/16" = 2.187"	2 7/32"	......	......	......
3/4" = .75" .	25/32"	2 1/16"	1 1/4"	1 3/16"	2 1/4" = 2.25" .	2 9/32"	4 5/8"	2 3/8"	2 1/16"
13/16" = .81" .	27/32"	......	......	......	2 3/8" = 2.37" .	2 13/32"	......	......	......
7/8" = .87" .	29/32"	......	......	......	2 1/2" = 2.5" ..	2 17/32"	......	......	......
15/16" = .93" .	31/32"	2 3/16"	1 1/4"	1 1/16"	2 5/8" = 2.62" .	2 21/32"	5 1/8"	2 1/2"	2 3/8"
1" = 1." ...	1 1/32"	......	......	......	2 3/4" = 2.75" .	2 25/32"	......	......	......
1 1/16" = 1.06" .	1 3/32"	......	......	......	2 7/8" = 2.87" .	3 1/32"	......	......	......
1 1/8" = 1.125"	1 5/32"	2 11/16"	1 1/4"	1 7/16"	3" = 3." ...	3 3/32"	5 5/8"	2 3/4"	2 5/8"
1 3/16" = 1.187"	1 7/32"	......	......	......	3 1/4" = 3.25" .	3 9/32"	......	......	......
1 1/4" = 1.25" .	1 9/32"	......	......	......	3 1/2" = 3.5" ..	3 17/32"	6 1/8"	2 3/4"	2 3/8"
1 5/16" = 1.31" .	1 3/8"	......	......	......	3 3/4" = 3.75" .	3 25/32"	......	......	......
1 3/8" = 1.37" .	1 7/16"	......	......	......	4" = 4." ...	4 1/32"	7"	3"	2 5/8"
1 7/16" = 1.43" .	1 15/32"	3 1/16"	1 1/2"	1 7/16"	4 1/8" = 4.125"	4 5/32"	......	......	......
1 1/2" = 1.5" ..	1 17/32"	......	......	......					

PUNCHES.



No. of Coupling.	Diameter of Punch at "X."								A.	B.	C.	D.	E.	Angle
B	1/8"	5/32"	3/16"	1/4"	7/32"	5/16"	11/32"	....	1 5/16"	11/32"	11/32"	1/8"	3/16"	30
	3/8"	13/32"	7/16"	....	....	....	....	....						
C	1/4"	9/32"	5/16"	11/32"	3/8"	13/32"	7/16"	....	1 3/16"	11/16"	21/32"	1/8"	1/8"	"
	15/32"	1/2"	17/32"	9/16"	19/32"	5/8"	....	....						
D	21/32"	3/4"	13/16"	....	....	....	....	....	1 3/8"	1 1/16"	27/32"	1/8"	11/16"	"
E	7/8"	15/16"	1"	1 1/16"	....	....	....	....	2 1/8"	1 1/16"	1 1/16"	3/16"	3/8"	"
F	1 1/8"	1 3/16"	1 1/4"	....	....	....	....	....	2 1/8"	1 5/16"	1 3/16"	3/16"	3/8"	"
G	1 5/16"	1 3/8"	1 7/16"	1 1/2"	1 9/16"	....	....	....	2 3/8"	1 13/16"	1 21/32"	3/16"	3/8"	"
H	1 5/8"	1 11/16"	1 3/4"	....	....	....	....	....	3 1/8"	2 1/16"	1 27/32"	3/16"	13/16"	"
I	1 13/16"	1 7/8"	1 15/16"	2"	2 1/16"	2 1/8"	2 3/16"	2 1/4"	3 5/8"	2 5/16"	2 7/32"	1/4"	1 1/4"	"
J	2 3/8"	2 1/2"	2 5/8"	....	....	....	....	....	4"	3 1/16"	2 27/32"	1/4"	1 3/8"	"
K	2 3/4"	2 7/8"	3"	3 1/4"	....	....	....	....	....	3 13/16"	3 7/32"	1/4"	1 1/4"	"
L	3 1/2"	3 3/4"	4"	4 1/2"	....	....	....	....	....	4 13/16"	4 7/32"	1/4"	1 3/8"	"

Fig. 96.

FIG. 97.

FIG. 98.

FIG. 99.

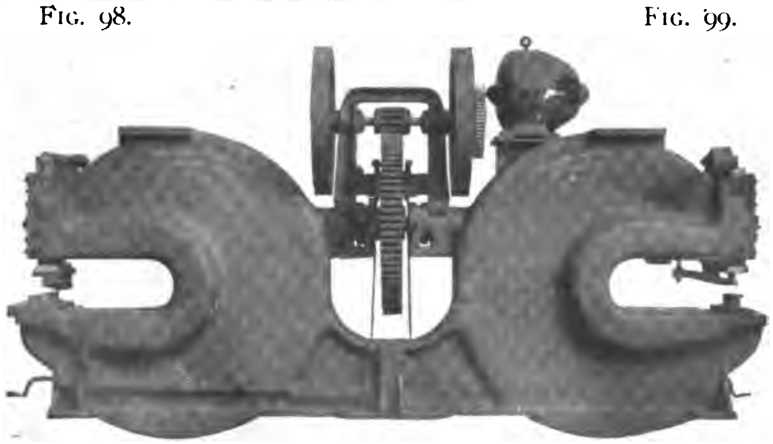
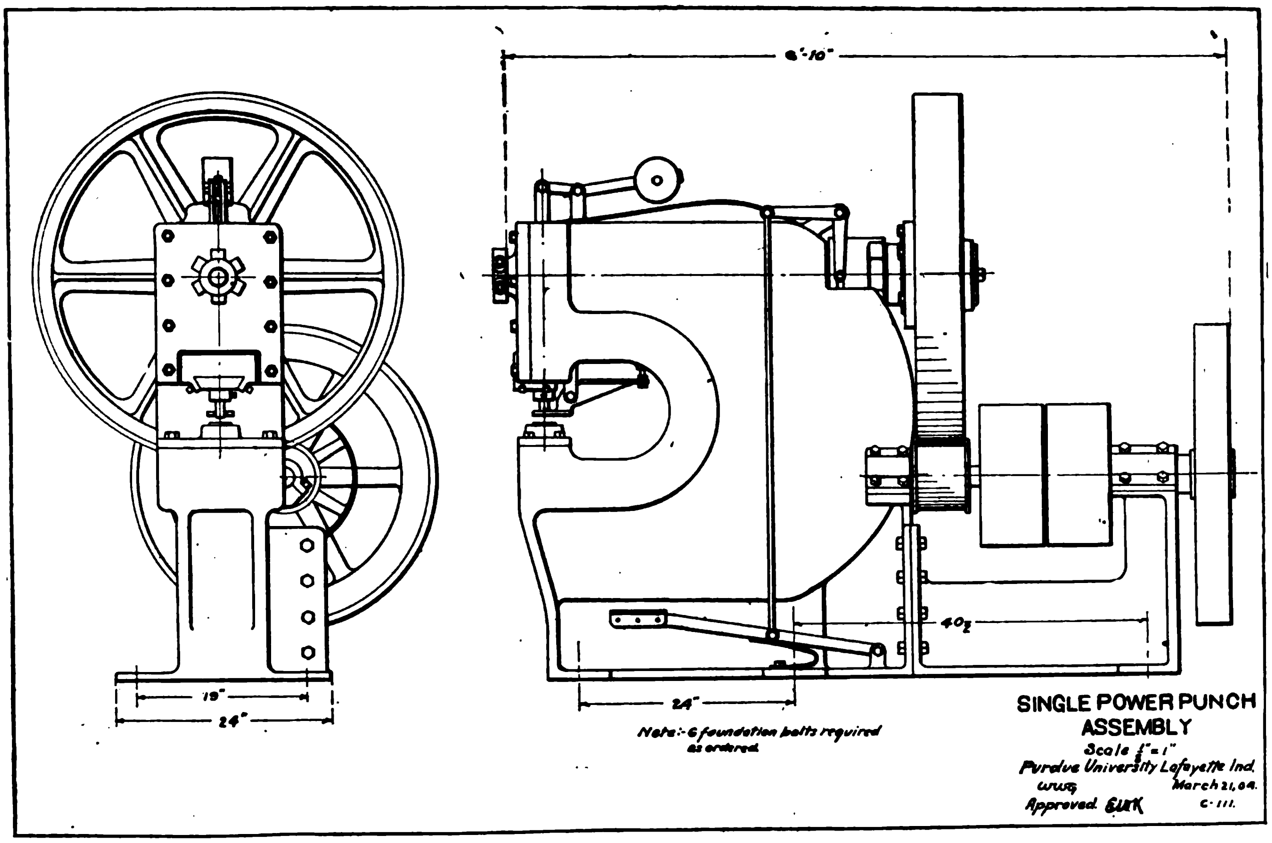
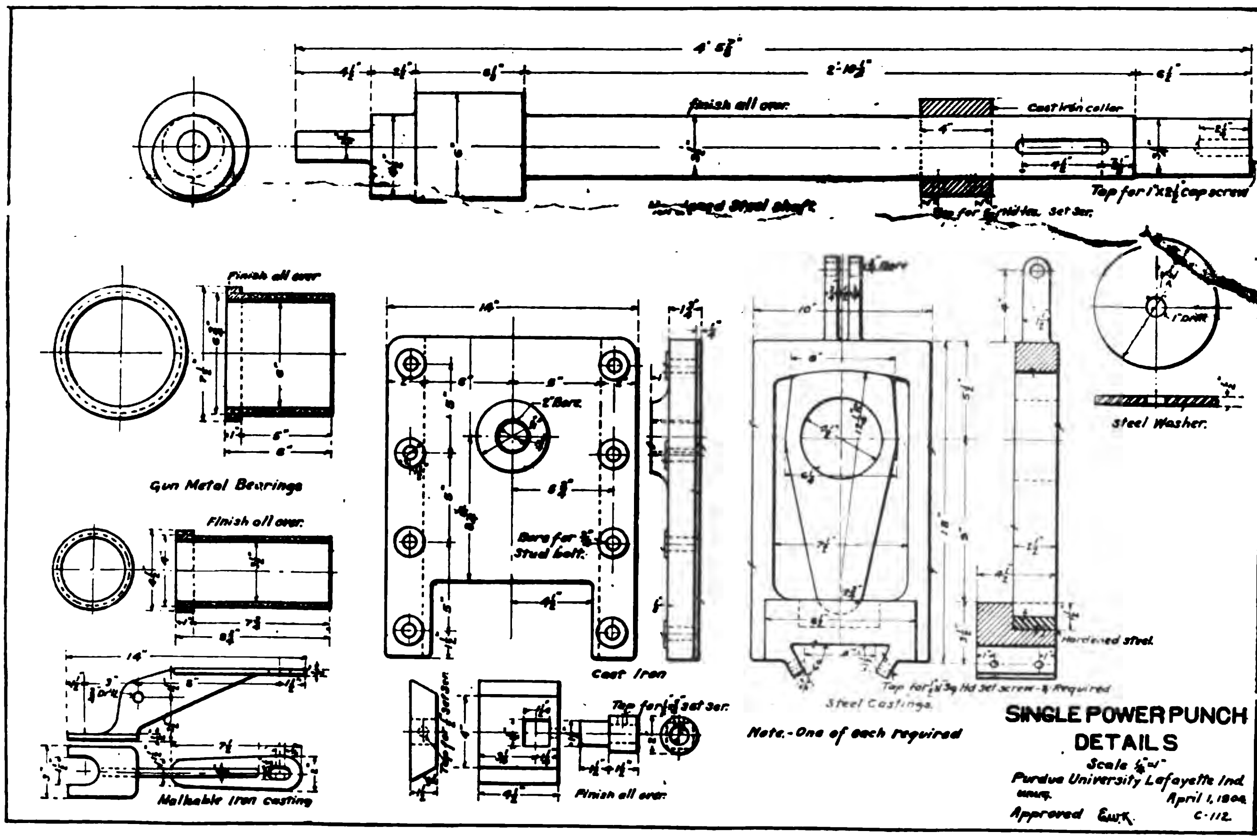
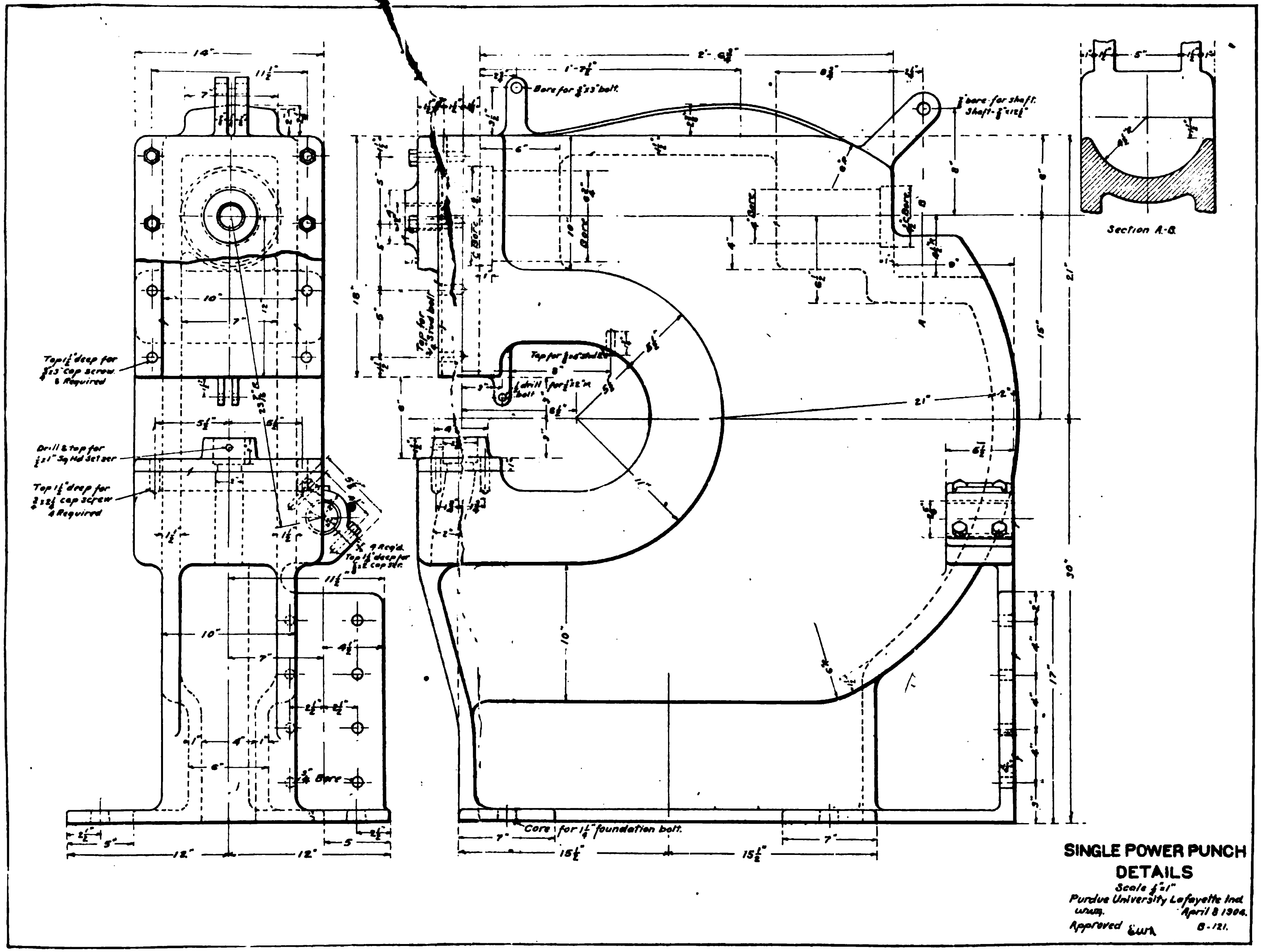
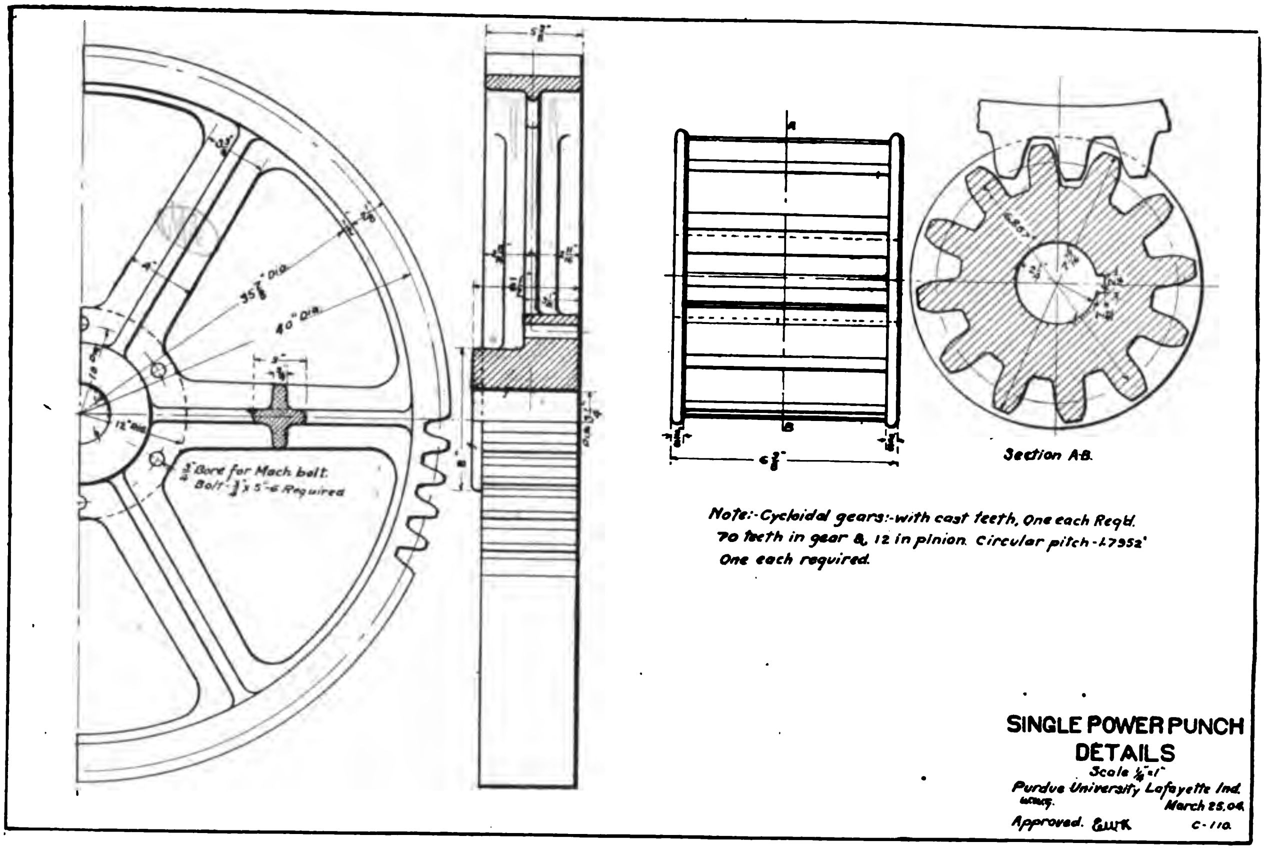


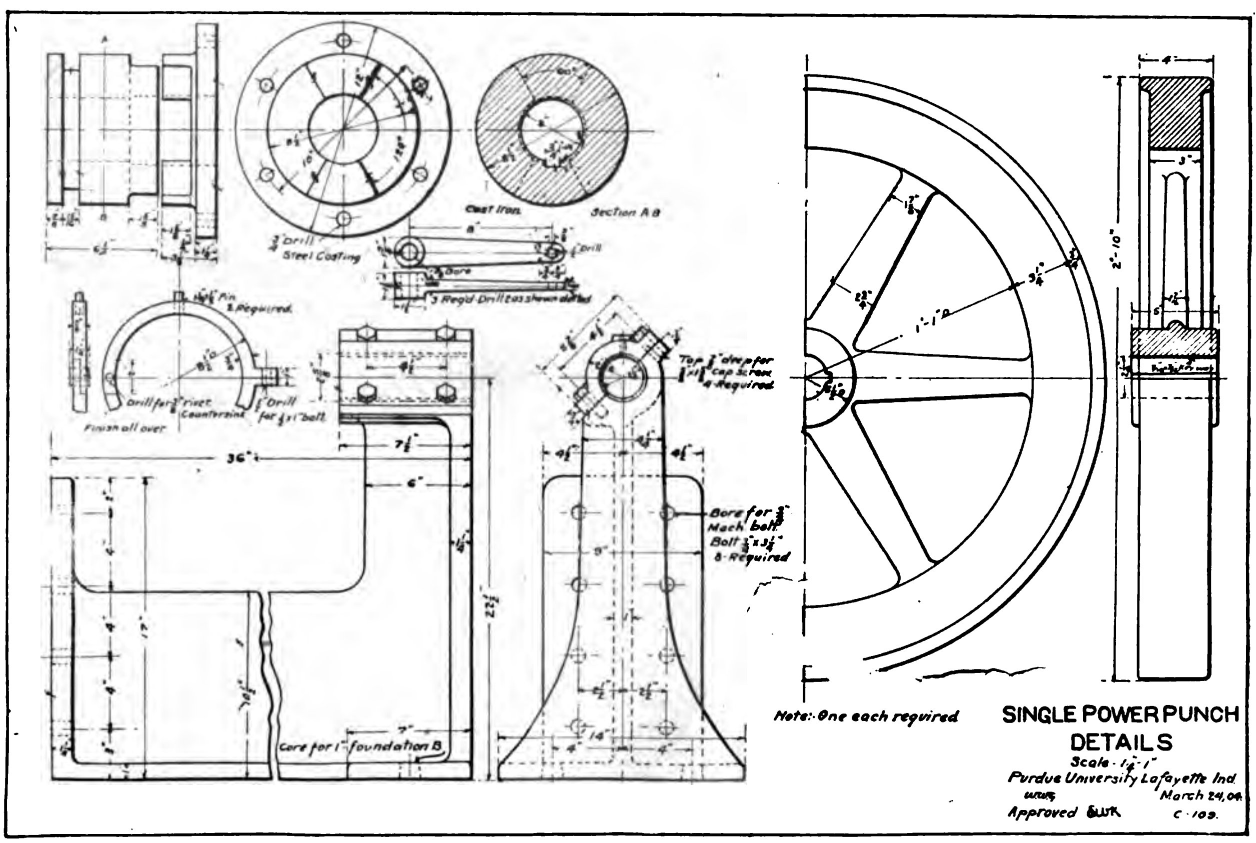
FIG. 100.


6'-10"
19"
24"
24"
40½
Note:- 6 foundation bolts required as ordered.
SINGLE POWER PUNCH
ASSEMBLY
Scale ⅛"=1"
Purdue University Lafayette Ind.
March 21, 04.
Approved
C-111.


finish all over
Cast iron collar
Tap for 1"x2½" cap screw
Steel Washer.
Gun Metal Bearings
Finish all over
Finish all over.
Cast Iron
Bore for Stud bolt.
Malleable Iron casting
Finish all over.
Steel Casting.
Hardened Steel
Note.- One of each required
SINGLE POWER PUNCH
DETAILS
Purdue University Lafayette Ind.
April 1, 1904.
Approved
C-112.

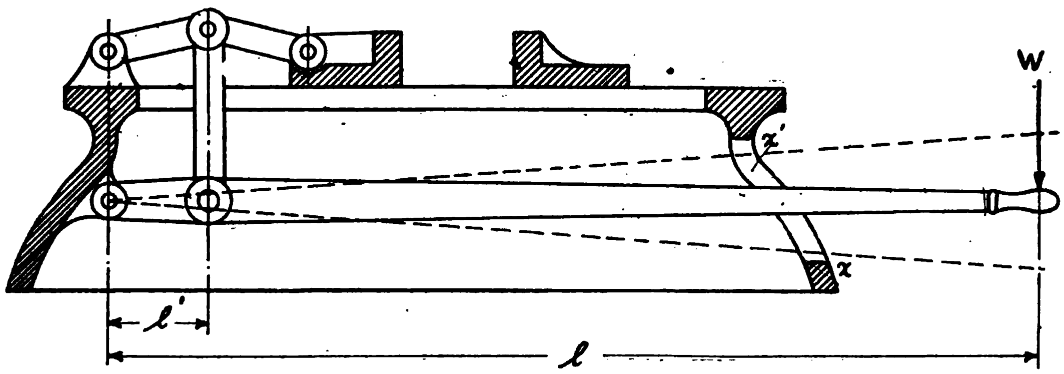

Section A-B
Bore for 5/8 x 3 bolt.
Core for 1 1/4" foundation bolt.
Drill & tap for 1/2 x 1" Sq. Hd. Set scr.
SINGLE POWER PUNCH
DETAILS
Purdue University Lafayette Ind.
April 8 1904.
Approved
B-121.


Note:- Cycloidal gears:- with cast teeth, One each Reqd.
70 teeth in gear & 12 in pinion. Circular pitch - 1.7952"
One each required.
Section A-B.
SINGLE POWER PUNCH
DETAILS
Scale 1/8"=1"
Purdue University Lafayette Ind.
March 25, 04.
Approved.
C-110.


Cast Iron.
Section A-B
Steel Casting
Finish all over
Core for 1" foundation B
Note:- One each required.
SINGLE POWER PUNCH
DETAILS
Purdue University Lafayette Ind.
March 24, 04.
Approved
C-109.

CHAPTER VII

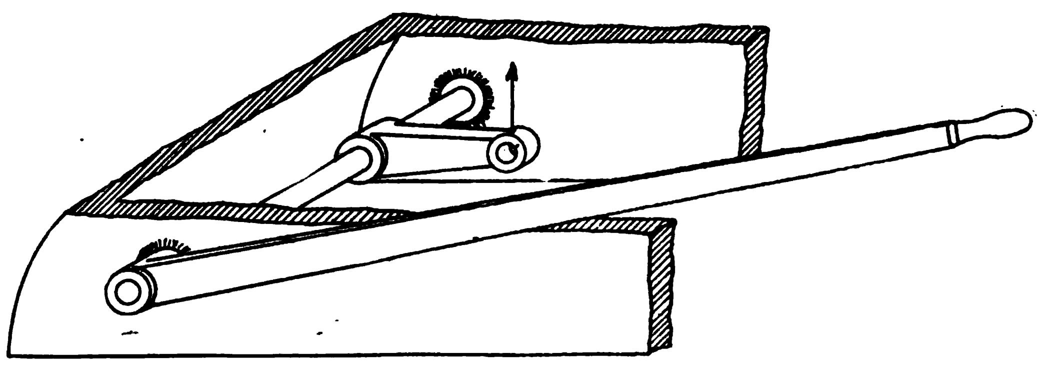
FIRST ALTERNATE. DESIGN NO. 1.



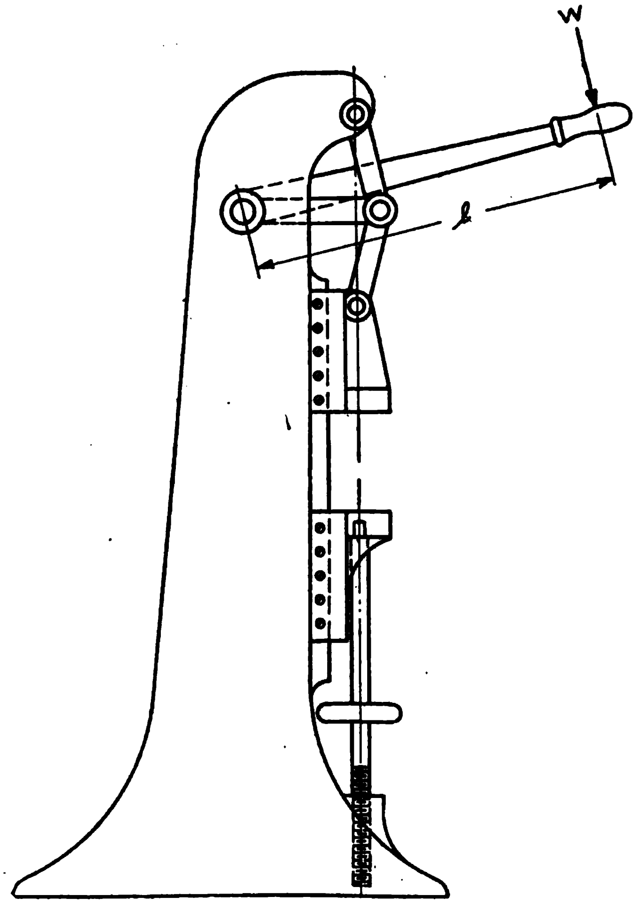
185. Assignment:—In this design the lever is placed within the bed rather than above it. It will be noticed that the end of the bed is slotted to allow for a movement of the lever arm between the points x and x'. The weakening of the bed due to this slot need not be considered a serious matter. With a long and shallow bed however, the movement of the arm will be small and will give a very slight movement to the sliding block. For our purpose this machine may be designed merely to exert a pressure between the two sliding blocks in which case a very slight movement is all that is necessary and the form shown will be satisfactory.

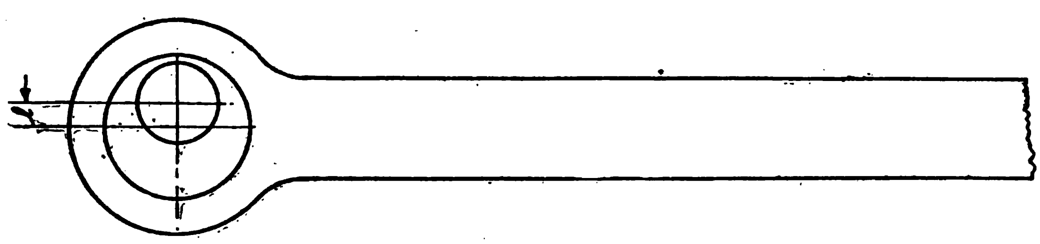
In case the movement of the sliding block is desired greater than that allowed here, the lever may be arranged as shown in the lower figure.

$W = \ldots.;\ l = \ldots.;\ l' = \ldots.;\ \theta = \ldots.$



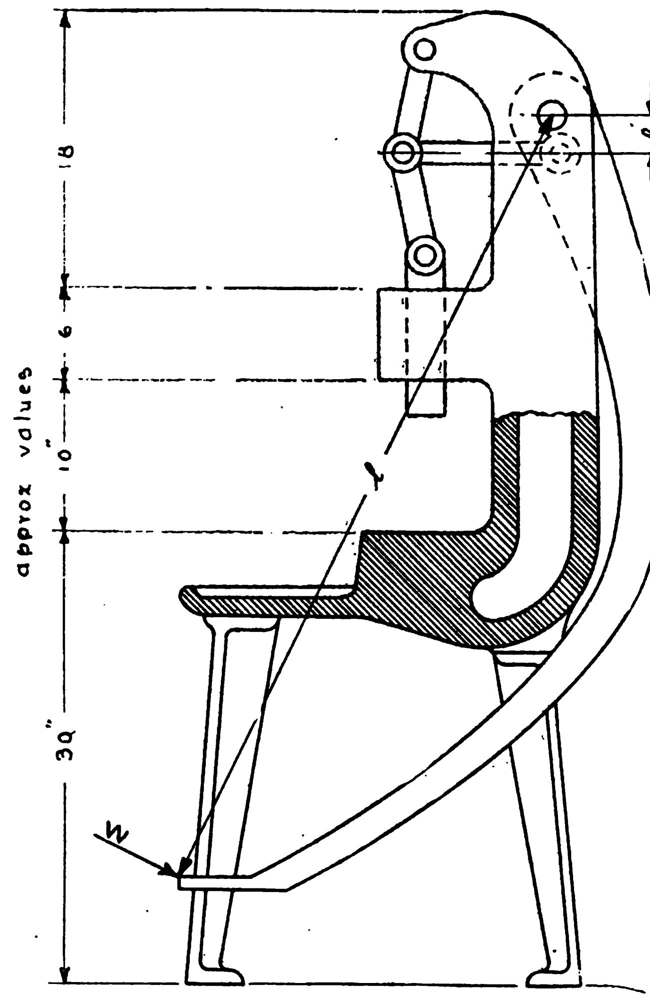
SECOND ALTERNATE, DESIGN NO. 1.





186. Assignment:—This design follows the principles laid down in No. 1, with two exceptions. First, the lever l' here becomes so small that a separate crank can not be used and an eccentric is substituted, the length of the lever l' being the distance between the centre of the shaft and the centre of the eccentric. Second, the thrust of the sliding block is received through a screw directly against the base of the frame. A hollow rectangular section is suggested as the best shape of the frame.

$$W = \ldots . \; l = \ldots . ; \;\; l' = \ldots . ; \theta = \ldots .$$



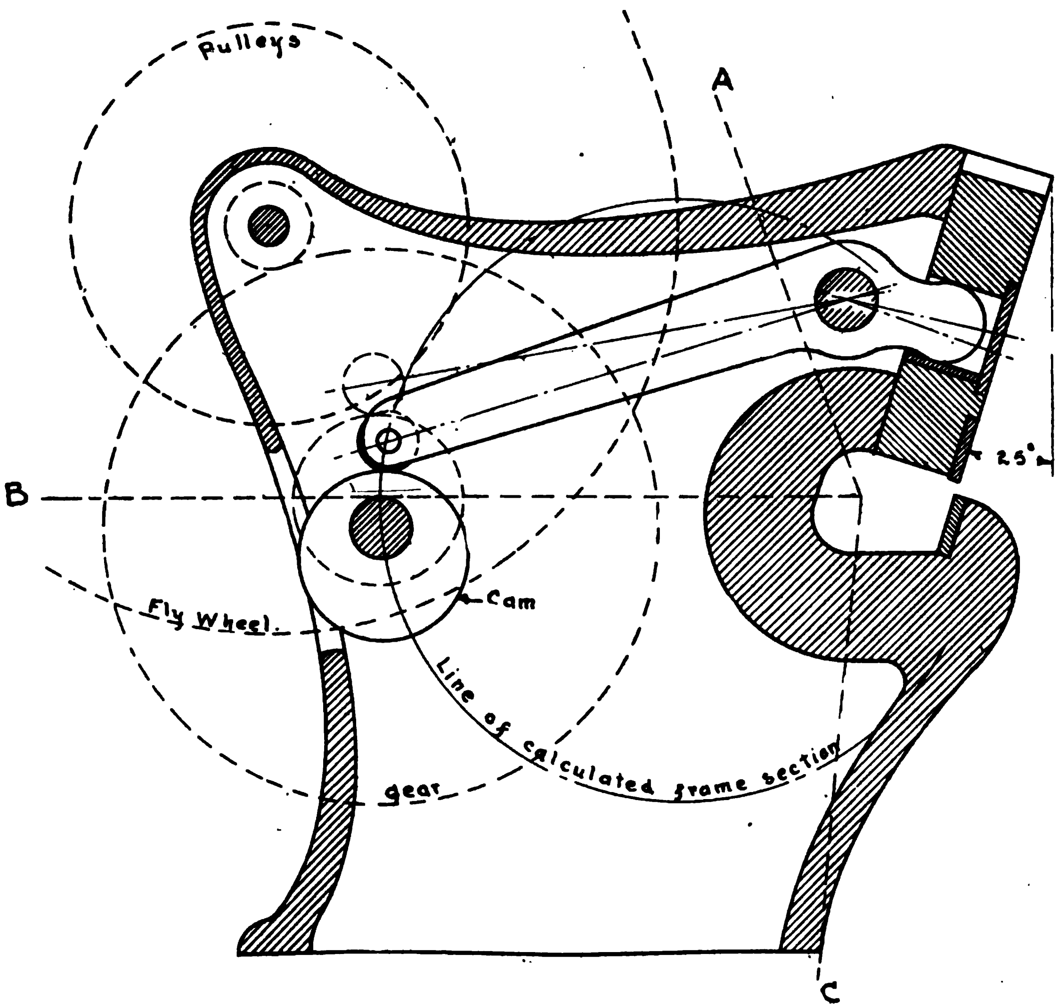
187. Assignment:—$W =$ (Not over 100#) $l =$ (60" to 72"); $l' =$ (3" to 6"); $\theta = \ldots\ldots$

This machine can be used for all kinds of light press work where but a small movement of the ram is needed. Where this movement is desired as great as possible, increase l' and decrease 1, also reduce the length of the toggle members.

The ram may be made rectangular in section and the forming dies need not be developed. The frame is hollow and the lever l is fastened on the plane of the toggle.

For illustration of a similar machine see 1900 catalog of the Niles Tool Works Company, Page 390.

FIRST ALTERNATE, DESIGN NO. 2.



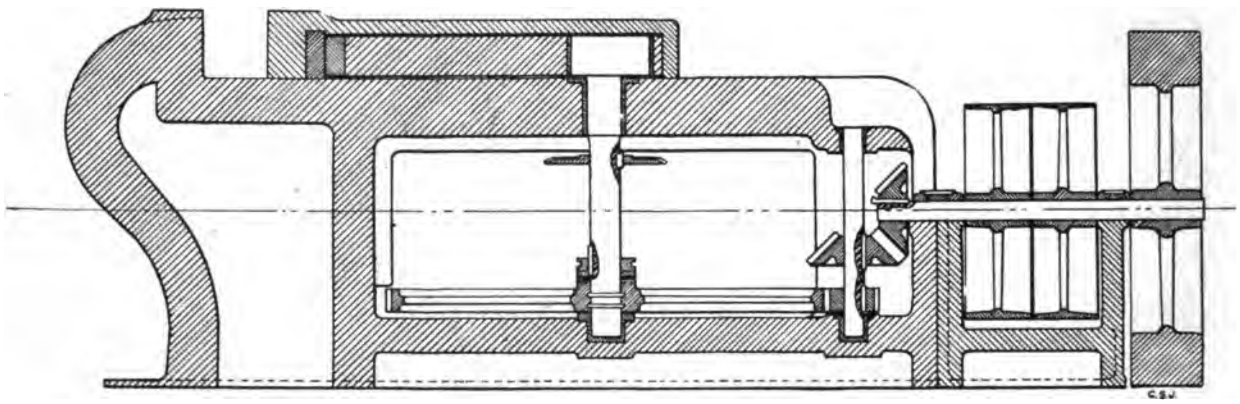
188. Assignment:—

Kind of material to be sheared ....
Width of plate to be sheared, ...
Thickness of plate to be sheared, ...
Greatest depth of throat, ...
Strokes of the ram per minute,....................................

The frame sections may be calculated if desired to a regular outline as shown in the dotted lines, after which modifications in this outline may be made by approximation. A better way, however, would be to sketch the approximate longitudinal frame section as above and figure for each of the several irregular sections, as *A*, *B* and *C*.

For description of machine and outside view see 1903 catalog of the Niles-Bement-Pond Company, Page 575.

SECOND ALTERNATE, DESIGN NO. 2.



189. Assignment:—

Kind of material to be punched..... ....

Size of largest hole punched, ... inches.

Thickness of the plate, ... inches.

Distance of centre of hole from edge of plate, ...inches.

Number of holes punched per minute................................

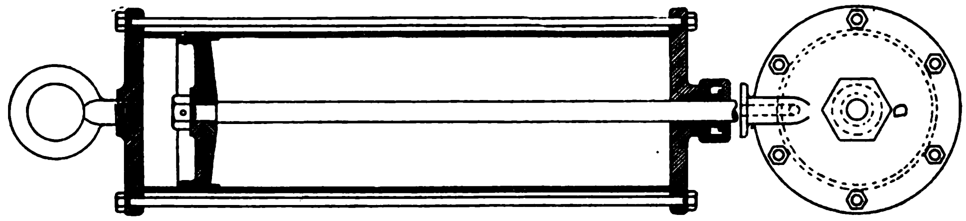
Horizontal punching machines may be designed in the same general way as the one described in the notes. It will be found that the frame sections may be calculated in the same way although the frame not being so regular will require a little more care in selecting the shaper and sizes of the various parts of the sections.

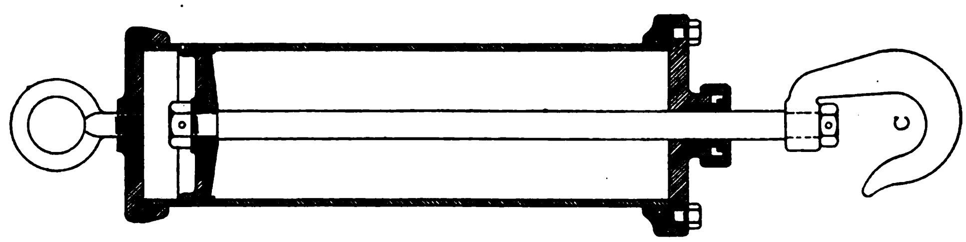
Machines of this type usually have a more shallow throat than the vertical type.

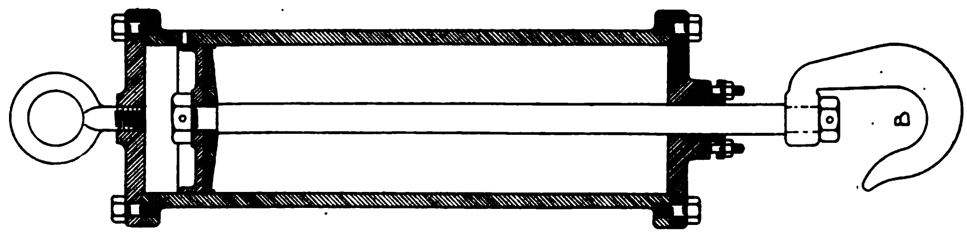
The line of the punch centre may be raised from the centre of the ram to the upper edge and is found convenient when punching near a shoulder.

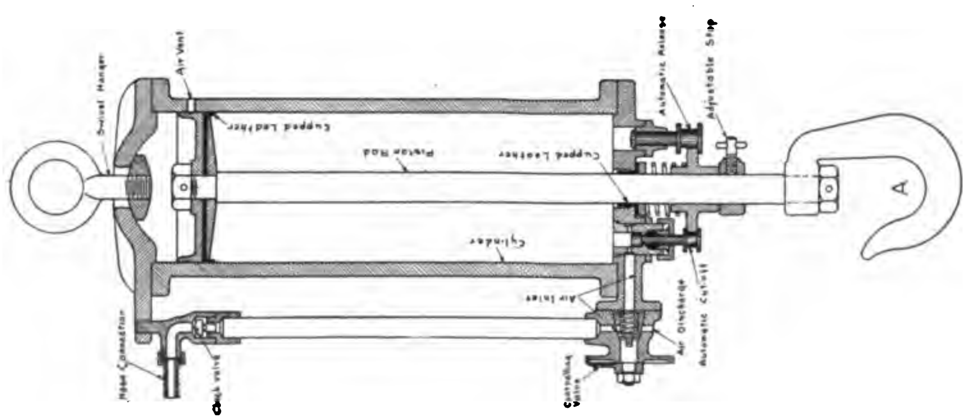
For description of machine and outside view see 1900 catalog of Niles Tool Works Company, Page 291.

AIR HOIST, DESIGN NO. 3.





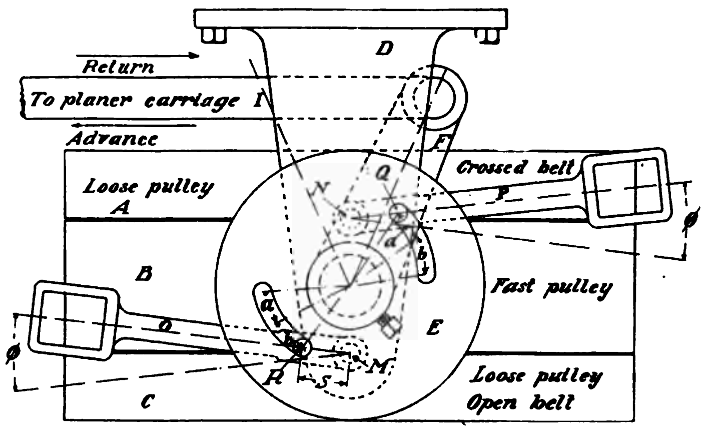




190. Assignment:—Capacity in free load............................ pounds.
Weight of parts and friction ....per cent.
Air pressure ....pounds per square inch.
Lift .... feet.

KINEMATIC SHEET No. 1.

Planer Cam.



a. *Arc of circle.*
b. *Arc of cam.*

191. Assignment:—

A and C are loose pulleys, B is a tight pulley.

D is fastened to frame of planer.

I moves back and forth, oscillating link F about G.

E is rigidly connected to F by set screw.

Levers O and P are pivoted at m and n on D.

Q and R are rollers fastened to the shifting levers.

Diameter pulleys .... Width of fast pulley. ....

Length of shifting levers.................. Width of loose pulleys, each. ....

$S =$ .... $\Phi =$ ....

Construct curve of cam so that the shifter will be constantly accelerated during first half of its motion and constantly retarded during latter half.

KINEMATIC SHEET No. 2.

Cam of Home Sewing Machine.

Diameter of Cylinder ....

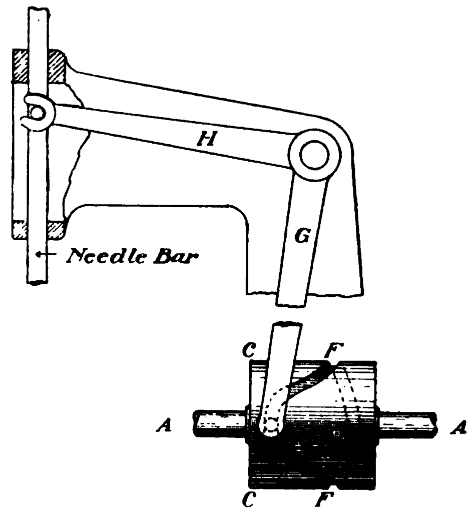
Depth of groove ....

Diameter of roller..............................

Stroke of bar ....

Length of arm G.............................

Length of arm H.............................

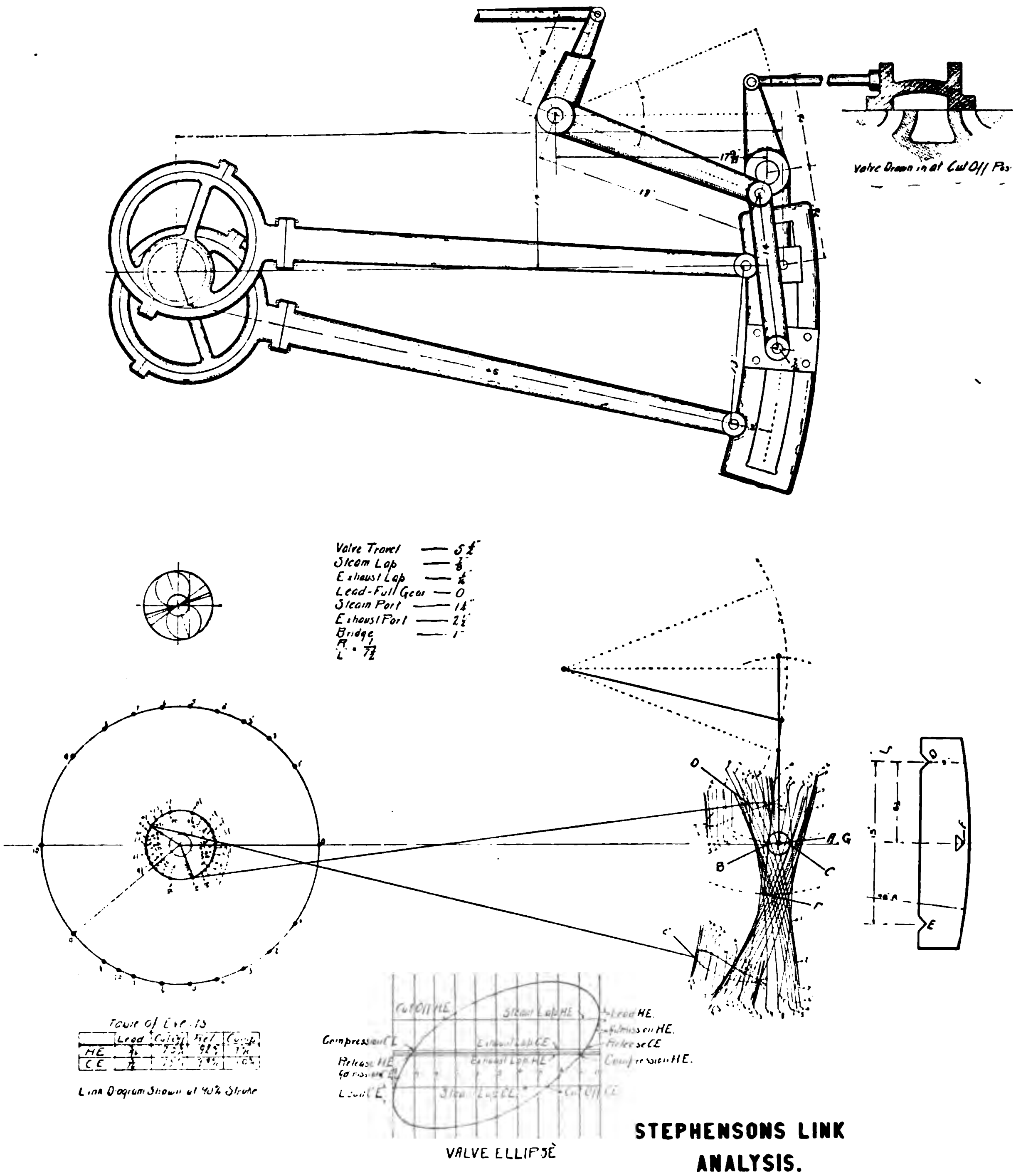


Drawn to Scale.

Design a cylindrical cam similar to that shown in the sketch to engage a rocker arm. Divide the motion into 24 time periods. The follower is to move with a constant acceleration during four time periods; during the next eight periods it is to move uniformily with the velocity attained; during the next four periods it is to come to rest with a constant retardation. The return motion consists of eight time periods; during the first four periods it is to be constantly accelerated and during the remaining four periods it is to be constantly retarded.

Required full projection of cam outline on the cylinder. This will require the development of the cylinder at top and bottom of groove.

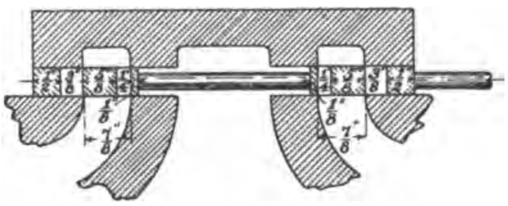
KINEMATIC SHEET No. 3.

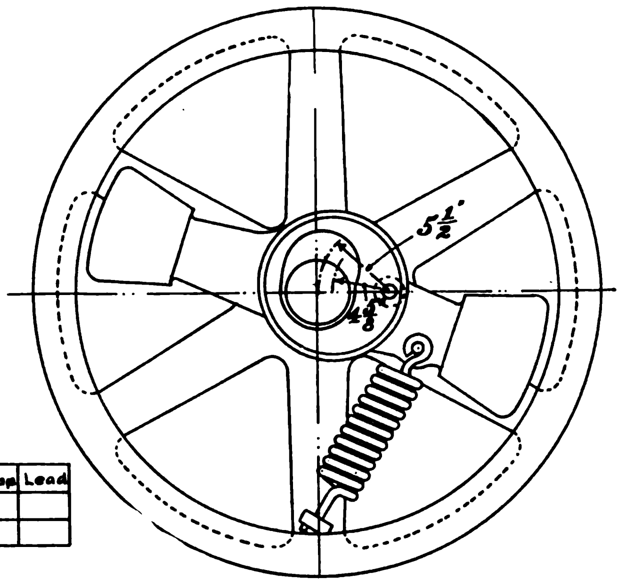


STEPHENSONS LINK ANALYSIS.

KINEMATIC SHEET No. 4.

Rites Inertia Governor.

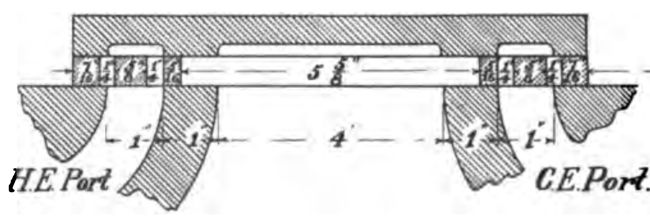


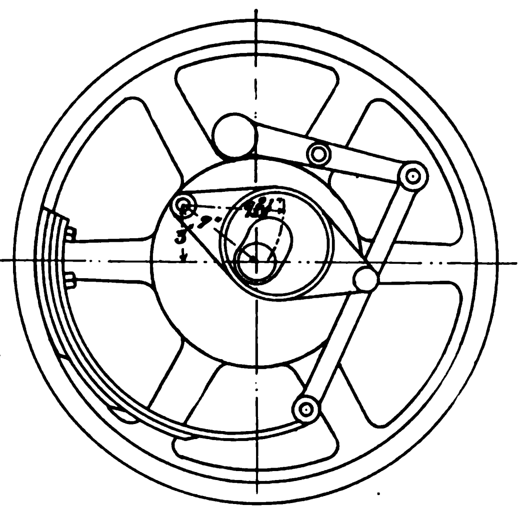


	Adm.	C.O.	Rel.	Comp.	Angle of Adv.	Valve Travel	Max. Port Opening	St. Lap	Ex. Lap	Lead
H.E.										
C.E.										

KINEMATIC SHEET No. 5.

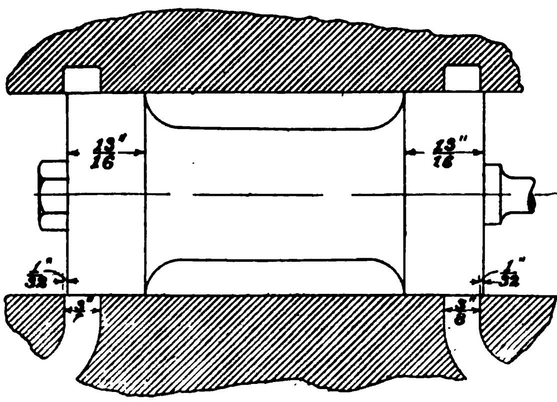
Straight-Line Governor.

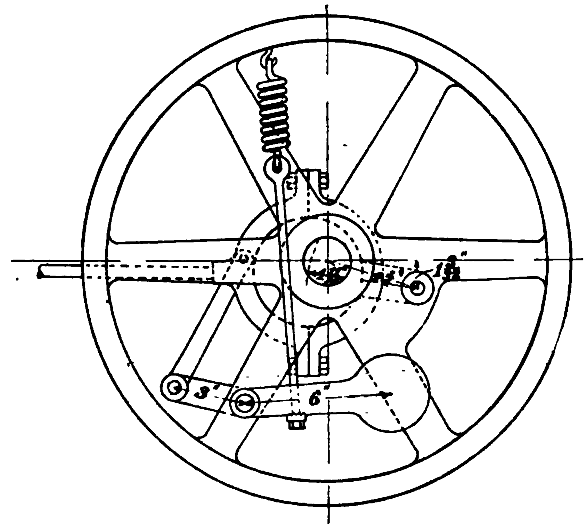




	Adm.	C.O.	Rel.	Comp.	Angle of Adv.	Valve Travel	Max. Port Opening	St. Lap	Ex. Lap	Lead.
H.E.										
C.E.										

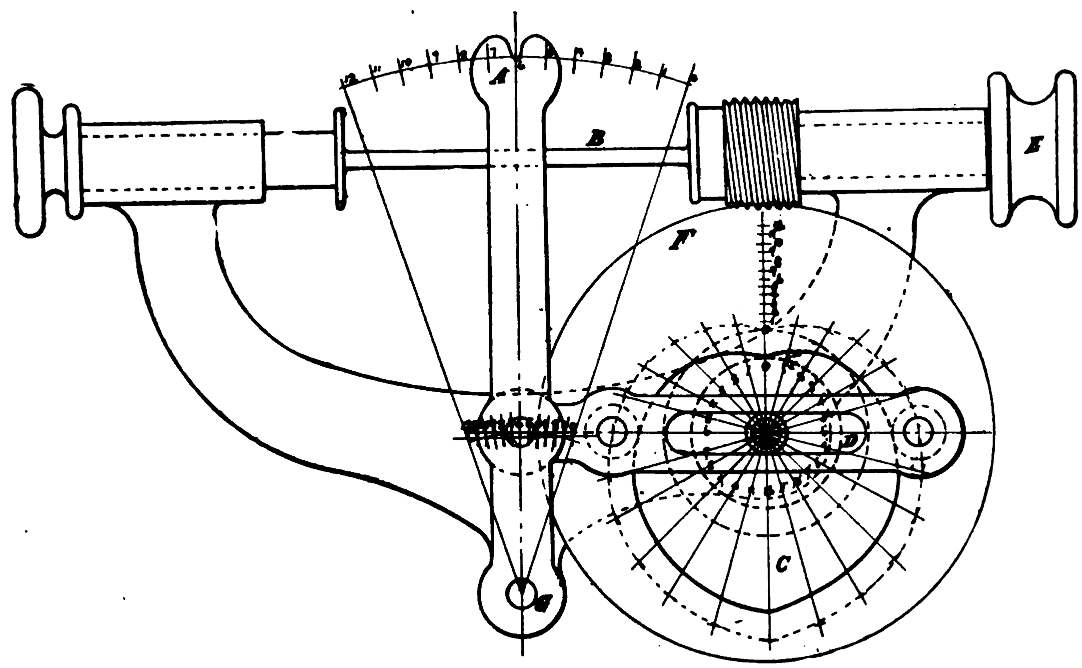
KINEMATIC SHEET No. 6.





	Adm	C.O	Rel	Comp	Angle of Adv.	Valve Travel	Max Port Open.	St. Lap	Ex Lap	Lead
H.E										
C.E										

KINEMATIC SHEET No. 7.



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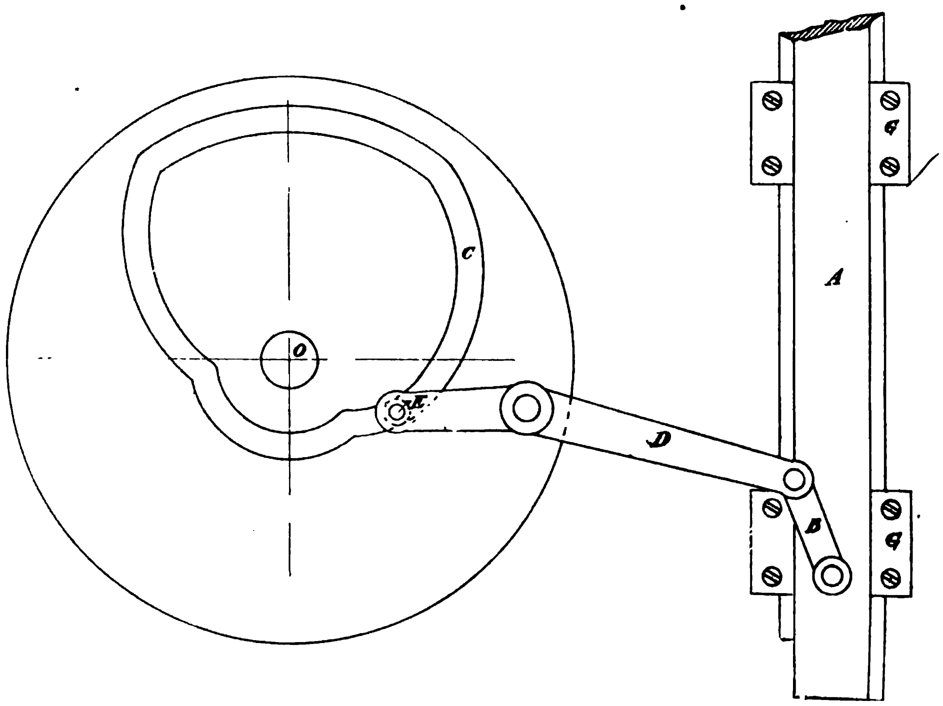

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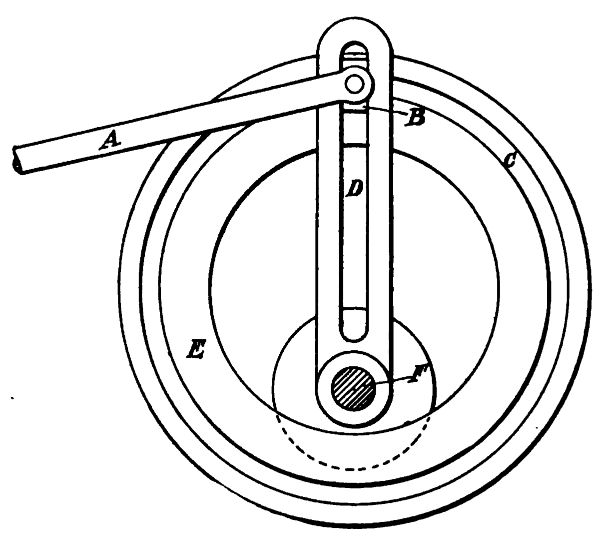
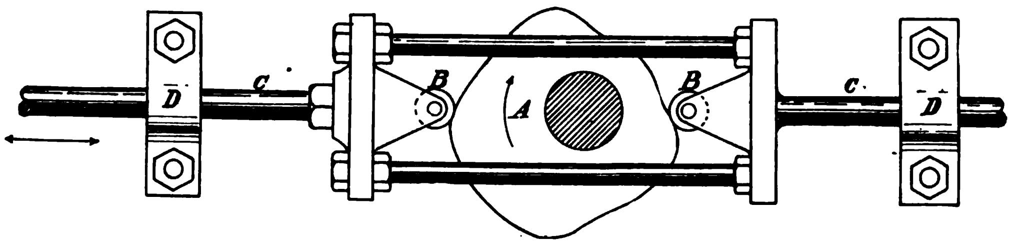
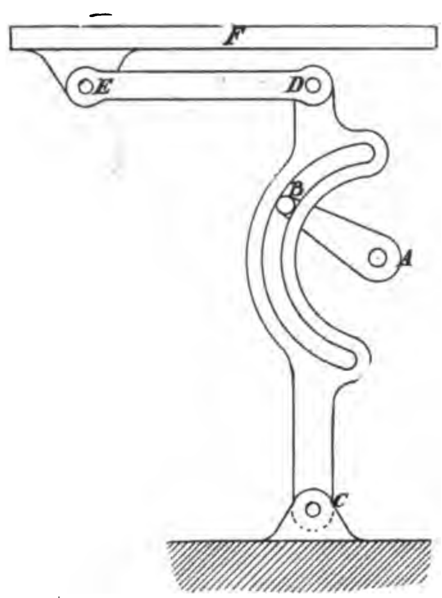
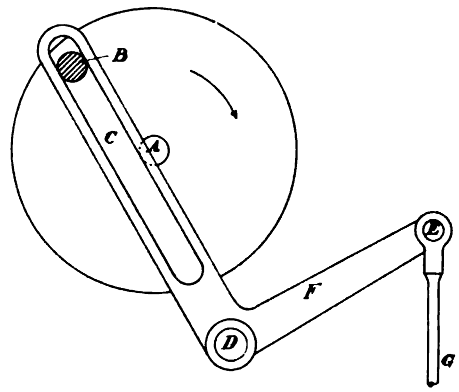


KINEMATIC SHEET No. 10.



KINEMATIC SHEET No. 11.



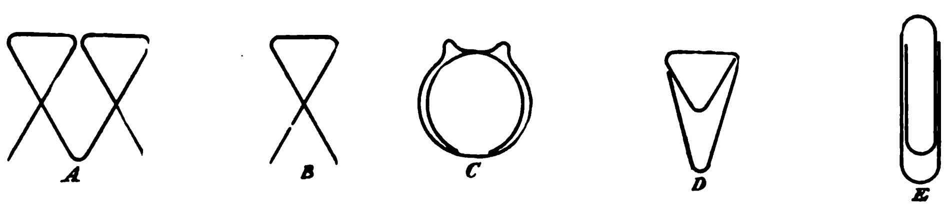


ORIGINAL PROBLEMS IN THE STUDY OF THE KINEMATICS OF MACHINES.

Illustrative problems, in the study of the mechanical movements of machines, are given in the American Machinist once each month, beginning Dec. 1, 1904. In order that original ideas may be developed, it is proposed that a line of bending or forming machines be designed. Attention will be paid exclusively to the *kinematics* of the machines and no parts will be figured for strength.

The following shapes of bent wires are suggested, from which a selection may be made. Any other machine, however, employing a system of cams may be used instead; as for example, a *bolt heading machine*.

KINEMATIC SHEET No. 12.



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